SEP 6 1939

Heating · Piping and Air Conditioning

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DUSTRY AND
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EMBER, 1939

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—Journal Section of
Societ of Heating
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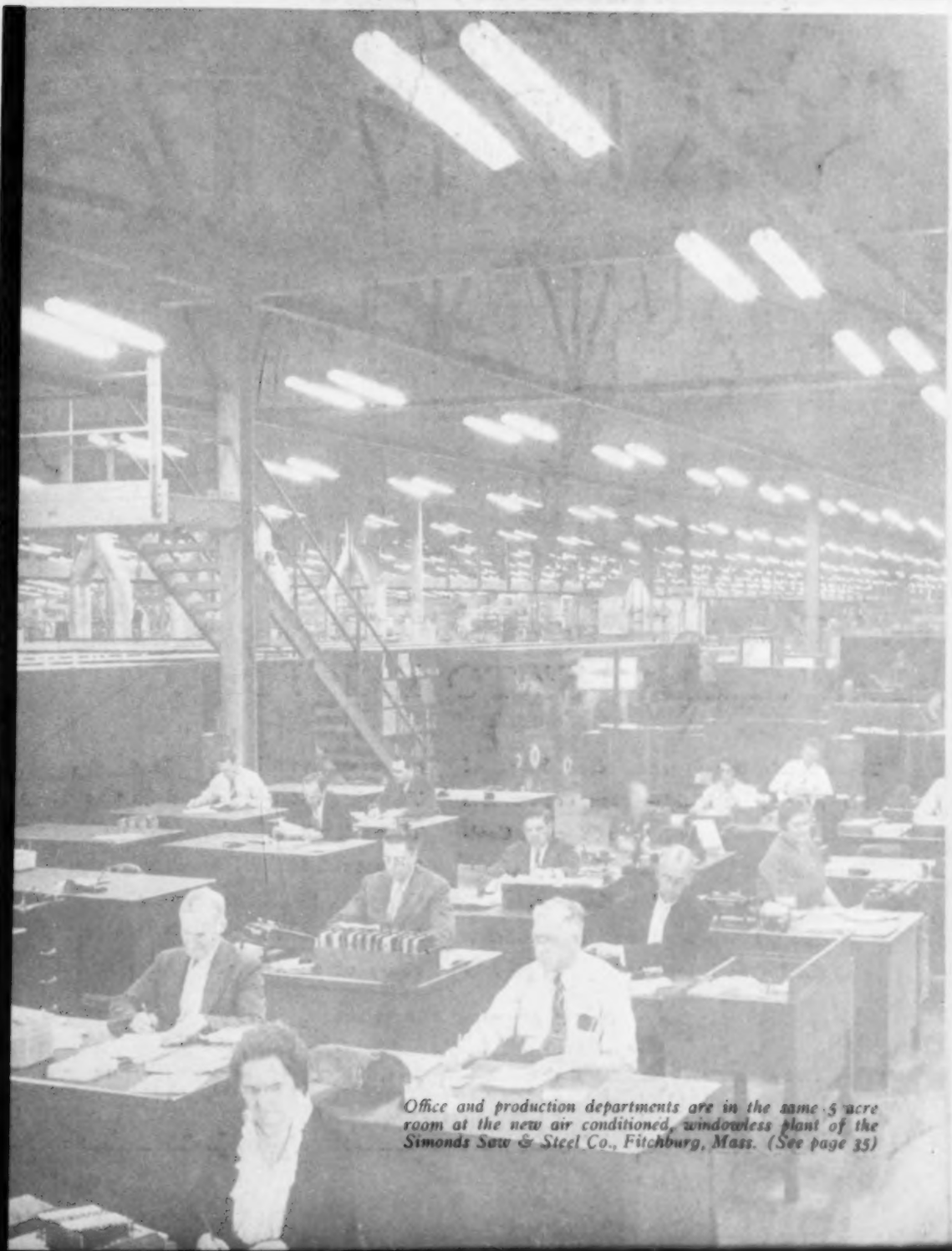

Office and production departments are in the same 5 acre room at the new air conditioned, windowless plant of the Simonds Saw & Steel Co., Fitchburg, Mass. *(See page 35)*

ARE YOU PAYING FOR HEAT AND BUYING A Shadow?
Condensed water collecting in unit heaters blocks off some of the heating surface – casts a long, cold "shadow" where there should be heat!

NO *"SHADOWS"* TO PAY FOR WITH THERMOLIER

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Grinnell's 50 years of heating experience created the Internal Cooling Leg . . . plus *13 other* outstanding points of superiority. They're all described in the 48-page Thermolier Data Book that covers Thermolier's 3 types and 30 sizes. Send for a copy. Grinnell Company, Inc., Executive Offices, Providence, Rhode Island, branch offices in principal cities of the United States and Canada.

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29% less steam consumed, Thermoliers went into a known textile plant.

Manufacturer with ove plants averages overall savin 30%. $50 per day in one alone, $5500 per year in one

GRINNELL THERMOLIER

THE UNIT HEATER WITH 14 POINTS OF SUPERIORITY

The Editor's Page

■ **The front cover** illustration (reproduced above) shows office employees of the Simonds Saw & Steel Co. working in the same 5 acre room where eight self-contained saw and file production lines are functioning. Production lines work from the rear of the plant toward finished stock bins which can be seen alongside the office area, the display lounge, and the engineering department. . . . This air conditioned, windowless plant at Fitchburg, Mass., is now in full operation; as described in the November, 1938, HPAC it was built in 1931 but because of the depression was not equipped or occupied until recently. . . . Four large air conditioning units of the spray type situated in four building extensions adjacent to the huge main room circulate 400,000 cfm of air through 3,000 lineal feet of overhead ducts, use 52 F water from four artesian wells yielding 1100 gpm. Air is also forced direct from the conditioning units to "man coolers" which moderate the temperature around the heat treating furnaces and drive warm air up into the curtained ceiling areas where all used air is exhausted through 42 roof ventilators. . . . Grinders and similar equipment are connected with 30 dust removal units having branches serving a number of machines. Air is filtered in these units and recirculated. Three flue gas removal systems take gases directly away from furnaces at 63 different locations. . . . Thirty-eight thousand feet of piping systems serve the plant. The lines for water, air, steam, gas, etc., come from overhead and branch down the column posts to the floor, fanning out under the floor to take care of the various machines. Control of the process piping for the individuals bays is centralized, making it possible in case of piping troubles to shut down only those machines directly affected. . . . Steam for processing and heating is provided at 100 lb pressure by three boilers of 225 rated hp each installed in a separate boiler house. . . . The structure is insulated against heat and vibration, and more than fourteen hundred 100 watt fluorescent tubes provide uniform lighting of 20 to 25 footcandles. . . . Engineers and builders were the Austin Co.

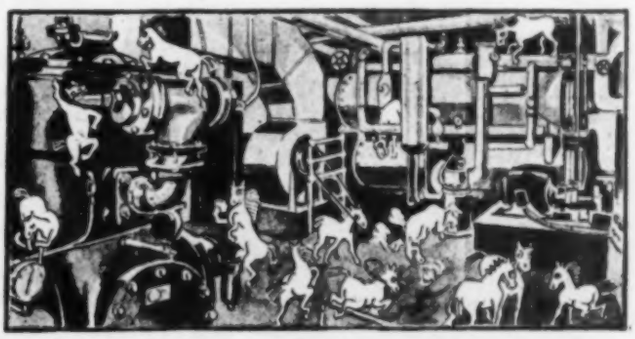
■ **Skytop, Pa.,** was the scene of a recent two day meeting of the Industrial Unit Heater Association which focused attention on the work the association has been doing, such as standardization of piping and standardization of motors. This latter has saved the industry thousands of dollars because the number of motors specified for use with unit heaters has been reduced from over 1000 to around 50. . . . Ninety-three per cent of all unit heaters sold are manufactured by members of the IUHA.

■ **Net profits** of Carrier Corp., air conditioning firm, for the first six months of 1939 were $137,742.41 in comparison with a loss of $332,697.52 for the corresponding period of 1938, according to an announcement last month by J. I. Lyle, president. . . . A contract for seven centrifugal refrigerating machines to supply cooling for the new Social Security Building now under construction in Washington, D. C., has been awarded to Carrier, according to E. T. Murphy, vice president in charge of marketing. Mr. Murphy said the installation was the "largest single refrigeration installation to be made at one time for a single building in the history of the air conditioning business." The seven machines will supply 4200 tons of cooling.

■ **The New York City** civil service commission, Municipal Building, Manhattan, is receiving applications for the position of assistant superintendent of school buildings, which is open to persons of all ages. . . . The commission is anxious to bring the examination to the attention of as many qualified people as possible.

This clever newspaper advertisement was used recently by Rothschild's —Kansas City, Mo., store—to feature its air conditioning system

ROTHSCHILD'S HAS THE CLOTHES



A Hot Tip on a Cool Subject from 255 Horses

Pardon us if we seem to brag a bit, but without a doubt we're the finest group of horses ever assembled by one owner. We boast sires by General Electric and Westinghouse and we graduated from Vilters condensing school. These famous names give us our famous pedigree.

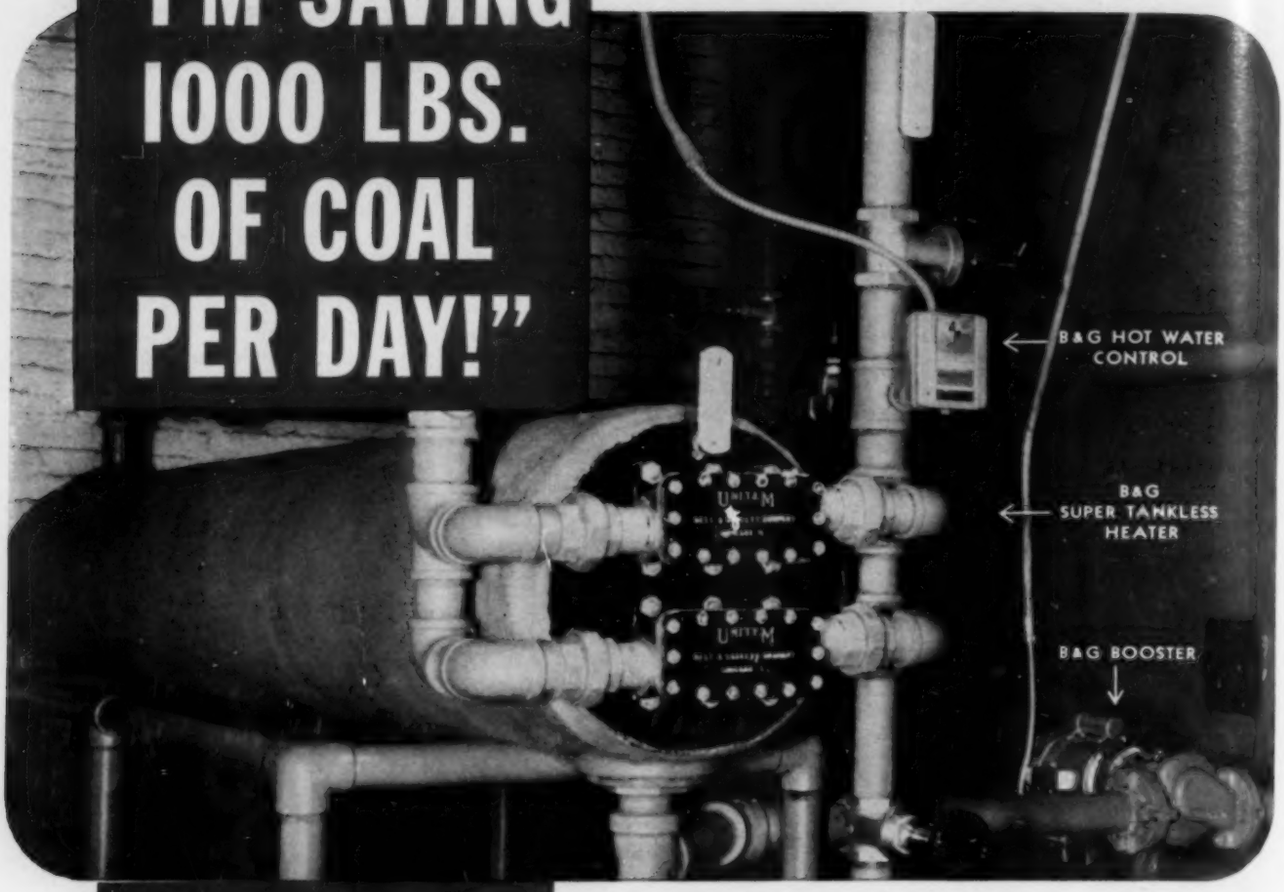
You see, we're the 255 thoroughbreds that run the finest air conditioning unit money can buy. We're "at home" 365 days every year. Two-thirds of that time we have fun just loafing, but, boy, the other third we really slave. Our work is hard, but we don't mind. We know that through our efforts every minute you spend at Rothschild's will be pleasant. Every nook and cranny is filled with washed and filtered air, delightfully cooled and refreshing. We make your shopping ever more pleasant at Rothschild's.

It's COOL at

Rothschild's

ON MAIN AT TENTH

"I'M SAVING 1000 LBS. OF COAL PER DAY!"



"B & G Super Tankless Heater reduced service water heating costs by nearly half" says building engineer

From 2400 lbs. down to 1400 lbs. of coal . . . a welcome saving of 1000 lbs. *per day* for this 168 apartment residential hotel. A good, but not unusual, example of the way B & G water heating equipment cuts expense corners.

In this building a B & G No. 17-24312 Super Tankless Heater replaced a steam coil installed in a storage tank. As shown in the illustration, it is a semi-tankless installation, with a 3" B & G Booster pump circulating boiler water through the heater. The pump is operated by a B & G Hot Water Control located in the hot water supply line. For circulating hot water through the building lines a separate B & G pump is employed.

This remarkable economy record should suggest a profitable approach to many similar buildings in your own business territory. If you haven't all the facts on B & G Water Heaters, write at once.

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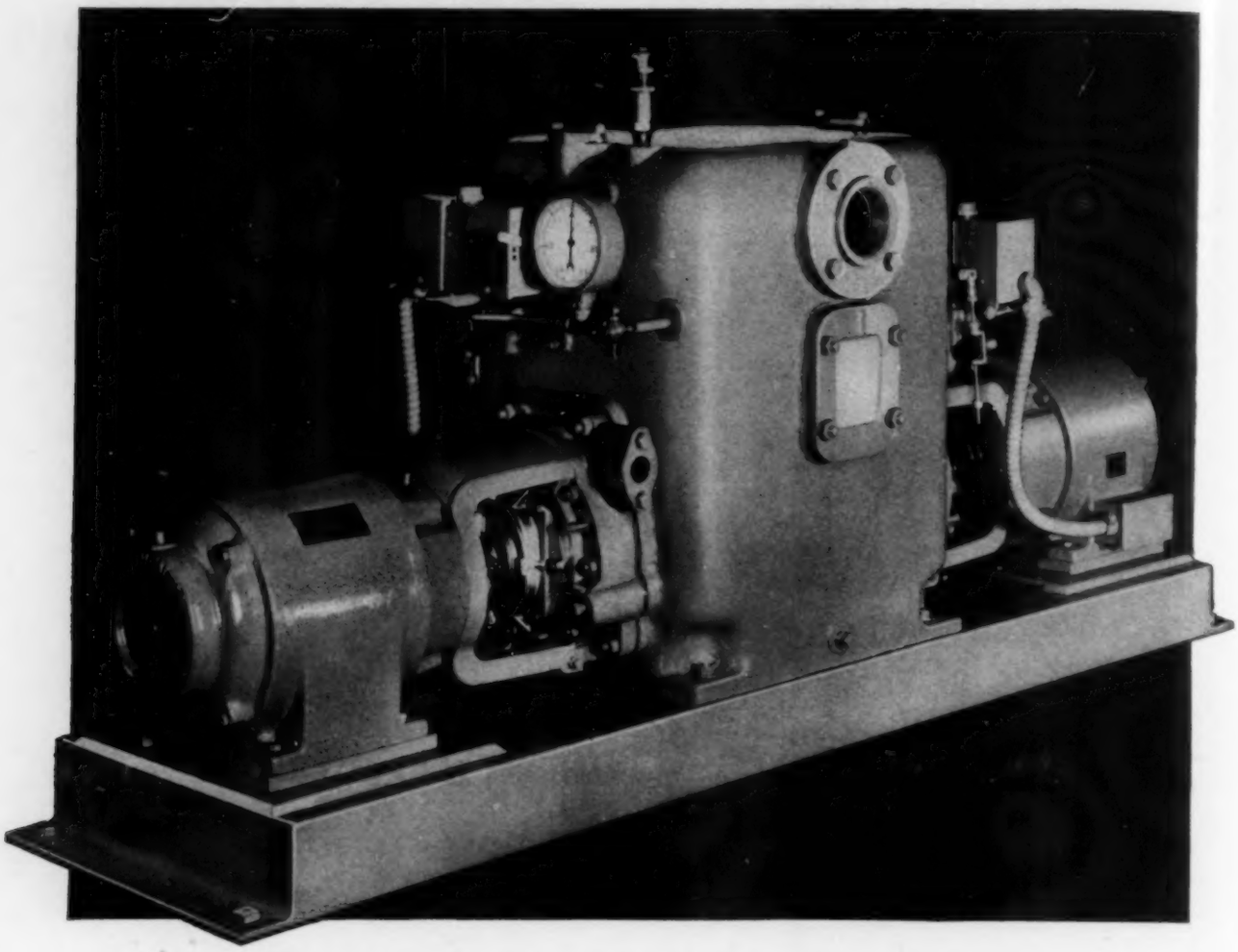

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Heating·Piping
and Air Conditioning

Vol. 11, No. 9 • *September, 1939*

Corrosion in Steam Heating Systems

Its Occurrence, Causes and Mitigation

By Leo F. Collins* with Everette L. Henderson†

This comprehensive treatment of the subject of corrosion in steam heating systems is based upon about 15 years of research work for The Detroit Edison Co. and is believed to be the most authoritative treatise on the subject that has ever been prepared. Part of the material was used in a thesis and term papers presented by Mr. Collins in partial fulfillment of the requirements for the degree in Master of Science in Chemistry at the University of Detroit. . . . It is felt that engineers concerned with operation and maintenance of building heating systems, district heating engineers, and industrial piping engineers, will find the information as useful as will designers and installers of heating plants. Because of its length it is necessary to publish this treatise in serial form. Accordingly, the reader is cautioned not to pass final judgment on any statement or data until the series is complete

TROUBLES resulting from corrosion of the internal surfaces of steam heating systems are usually manifested as the accumulation of insoluble products at restricted points, and as leaks at the threaded ends of small lines. Some regard such troubles as inconsequential, while others believe them to be of considerable moment. Between these two extremes, there are many who hold that while corrosion accounts for serious financial losses only in extreme cases it does constitute a general nuisance and should be prevented if possible.

As an employee of The Detroit Edison Co. since 1925, the senior author has devoted much time to a study of various aspects of the subject. This is especially true during the time he was simultaneously a graduate student. The studies are still being pursued because satisfactory answers to the many ramifications of the problem have by no means been obtained. On the other hand, the data collected reveal some facts which should be of general interest and use.

The data were all obtained in plants of The Detroit Edison Co. or in Detroit buildings. From a geographical standpoint, therefore, the scope of the studies is somewhat limited. Even so it is felt the information obtained is of such a fundamental nature as to warrant the following general conclusions:

1) Corrosion in a steam heating system cannot be absolutely prevented, no matter how much diligence is exercised. Even a negligible amount of corrosion may provide a sufficient amount of insoluble material to accumulate at certain points and thus restrict or stop the flow of condensate. Annoyances of this nature, therefore, may be expected occasionally even though the steam used is of the best quality (chemically) that can be produced commercially.

2) Carbon dioxide alone or associated with oxygen, when *dissolved in condensates*, are the *basic* causes for practically all corrosion troubles in ferrous metal lines.

3) Carbonic acid causes most rapid corrosion and accounts for many pipe failures.

4) Oxygen is usually responsible for pitting, and for the production of insoluble materials which collect at restricted points.

5) The principal source of carbon dioxide is entrainment with the steam, while infiltration of air into a heating system usually is necessary to cause appreciable amounts of oxygen to dissolve in the condensates.

6) Piping materials are available which are quite resistant to both oxygen and carbonic acid attack. Their use, however, is usually limited, for either mechanical or economic reasons, to rather special services.

7) No corrosion inhibitor is known to the authors which they feel can be recommended *unqualifiedly* for addition to steam.

The studies which form the basis of this presentation have embodied the following research work:

1) Experimentation with four methods for measuring corrosion in operating systems and the final development of a short time indicator of corrosion rates which gives reliable results when intelligently used.

2) The development of procedures for collecting samples of steam and condensate from various points in operating systems and the adaption of analytical procedures to meet the peculiar requirements of such studies.

3) The measurement of corrosion under operating conditions and at different points in the systems of about 25 buildings.

4) Laboratory measurements of corrosion under conditions designed to reflect the influence of only gases entrained with steams.

5) The analysis of steams developed in 12 boiler plants.

6) A few experiments to study the feasibility of adding inhibitors (oil, sodium silicate, and ammonia) to steam.

*Chemist, The Detroit Edison Co.
†Professor of Chemistry, University of Detroit.
Part 1.

7) The installation of 440 weighed specimens (each 1 ft in length) of 11 different types of pipe in the return lines of 11 different buildings, and of 35 specimens of 12 different types in the return lines from five different water heaters, to determine the comparative corrosion resisting abilities of the more common piping materials.

8) Laboratory and plant experiments to determine the possibility of producing commercially non-corrosive steams from feedwaters saturated with air and containing different quantities of carbon dioxide, when the latter is present either as a free gas or in chemical combination.

Scheme of Presentation

The studies indicated under items 4 to 8 inclusive are not finished. In the case of those outlined under item 7 the results from only one building return line are now complete. Those indicated under items 1 and 2 are not reviewed because of their limited interest. The results of the other studies are presented in four parts: The first part discloses the data collected from various building heating systems. In the second part an effort is made to demonstrate, from these data, and from those obtained as outlined under items 4 and 5, the factors primarily responsible for corrosion. The third part reviews, in a detailed manner, the studies made to discover ways for ameliorating corrosion troubles. The fourth part describes a water treating method used in a large central heating plant to produce a steam containing an average of about 3 ppm [parts per million] of CO_2 and no oxygen from a feedwater initially saturated with air and containing up to 95 ppm of CO_2, most of which is initially present as alkaline earth bicarbonates.

Throughout this paper corrosion rates are reported in absolute values and whole numbers. These values have been obtained by determining the losses in weight of the specimens for known time intervals and calculating these as the equivalent average penetration in inches per year. The results of such calculations are multiplied by 1000 to facilitate ease in reporting.

A calibrated corrosion rate scale also is referred to in certain parts of the discussion. In this, values up to 10 represent "negligible" corrosion, those between 10 and 20 "mild" and those above 20 "active." These limits were arrived at by measuring corrosion under fairly well known conditions which permit of such an evaluation. While they are approximate, it is significant that in checking the scale against actual corrosion troubles (over a period of about four years) it has not been found in serious error.

Occurrence of Corrosion in Operating Steam Heating Systems

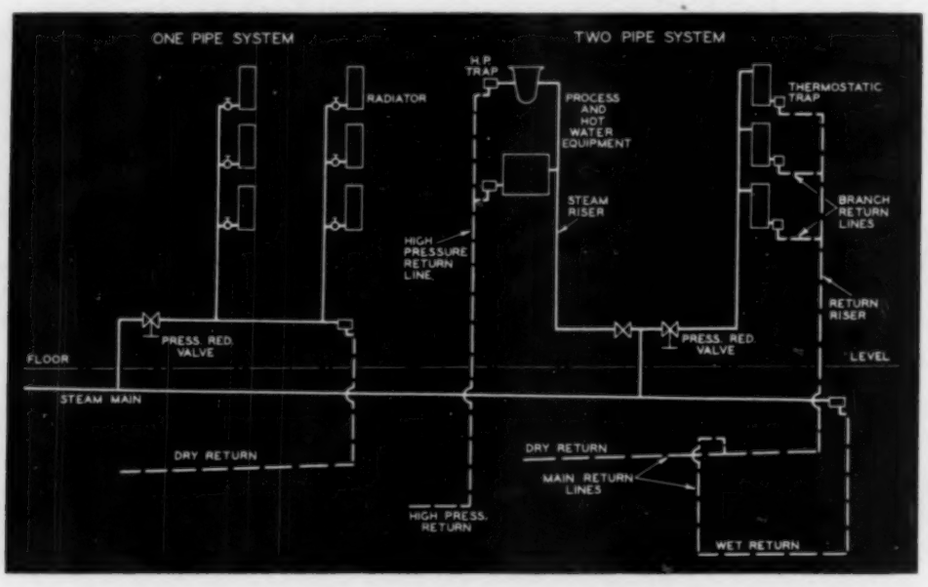
When corrosion troubles are encountered it is almost inevitable that three questions will be asked: Why did it occur? Why did it occur at the particular location in the system? Why do not similar troubles occur in other systems using the same steam or in duplicate systems using steams of comparable chemical quality?

In answering such queries many physical as well as chemical factors must be properly evaluated. The former obviously involve considerations identified with the design and operation of heating systems. It is not within the capabilities of the authors to describe fully all of those peculiarities which may be influential. As a basis for discussion it should be appropriate, however, to point out the integral parts which are involved.

Arrangement of Rudimentary Heating Systems

For the purpose at hand it will suffice to classify steam heating systems as of either the one pipe or two pipe type. The essential parts of each are shown schematically in Fig. 1. It can be seen that some lines transport only steam, that the purpose of other parts is to condense the steam, and that the condensate formed is subsequently carried away by branch return lines, return risers, dry return, and in some few cases wet return

Fig. 1—Typical arrangement of building steam heating systems



lines. In the case of one pipe systems the same pipe performs the dual function of transmitting both steam and condensate, flowing counter current to each other.

Heating systems may be operated either continuously or intermittently, under positive pressure or under vacuums. As a result the exposed surfaces may experience repeated changes, such as temperature fluctuations, alternate wetting and drying, expansion and contraction—which conceivably are conducive to corrosion. When these studies were undertaken it was not known whether other variables might not also be influential. Nor was it at all certain what effect various possible combinations of those known might produce. Because of these circumstances it was reasoned that it would be foolhardy to attempt to duplicate operating conditions in laboratory experiments; and that a more reasonable approach would be to investigate a representative number of operating systems. This, it was hoped, would disclose not only the important factors but also give an index of the influence of each. For about three years, therefore, the major part of the work consisted in making corrosion measurements in operating systems in an effort to obtain such information.

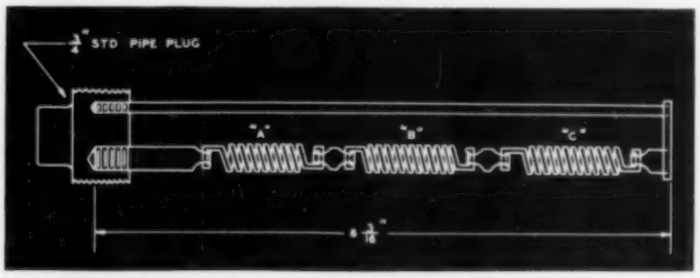


Fig. 2—The NDHA corrosion tester

The Corrosion Measurements

Thus far the rate of corrosion has been measured (at one or more points) in the heating systems of about 25 buildings. In some 12 additional buildings other relevant data have been obtained.

In measuring corrosion, the tester developed by the corrosion committee of the National District Heating Association was used.[1] This device, Fig. 2, utilizes three coils of steel wire as the test specimens. Each coil of wire is weighed on an analytical balance, after which it is assembled in the retaining frame. Following exposure the products of corrosion are stripped off chemically and the specimens are reweighed.

Since the wire used in making the helical coils is of uniform diameter (0.05 in. ± 0.0005 in.) and density, and since the specimens are exposed for a known period and the loss in weight is measured, it is possible to calculate the area exposed, the volume of metal lost, and therefore the average rate of penetration. This is conventionally reported in inches per 1000 years.

The coils are insulated from one another and from the supporting frame to preclude the possibility of electrolytic action. A special wire of uniform chemical composition and physical characteristics is used in making the helical coils. It is at least comparable to the kind of metal used in fabricating much of the ordinary grade of steel pipe used in heating systems.

Conditions Under Which Measurements Were Made in Operating Systems—Measurements were made by subjecting testers to the following conditions:

1) Immersed in condensates produced by bleeding steams from continuously operating mains through air cooled condensers. (The conditions thus produced are comparable with those which exist in steam mains continuously under pressure.)

2) Immersed in drips (and in some cases, condensate produced as in item 1) from building steam risers in both pressure and vacuum heating systems.

3) Immersed in condensate in the dry return lines of systems operating under pressure or vacuum.

4) Immersed in condensate in the wet return lines of systems operating under pressure or vacuum.

5) Inserted in condensing equipment; i.e. in the bottom of operating radiators in pressure systems and in the condensate lines from water heaters.

Results Obtained—The results obtained are shown in Fig. 3. This chart is informative when correctly interpreted but it is singularly misleading when improperly evaluated. The following discussions disclose both possibilities.

It is evident that at the locations studied "active" corrosion was found in only a few instances. It is obvious, also, that for equipment of a given type (as, for example, in dry return lines) the rate of corrosion was quite comparable at all locations. Since the steams used in these buildings were developed at 12 different sources, four of which were district heating plants and the remainder isolated boiler plants, it could be concluded that the chemical quality of steam has little bearing upon the corrosion problem; that in these buildings corrosion actually presents no problem.

In contradistinction, it is to be noted that few measurements were made in condensing equipment, that at *all* locations some corrosion was found, even though in the majority of cases it was of the "mild" variety.

With respect to the measurements made in steam lines and steam risers it should be pointed out that they reflect the rate of attack suffered by such operating lines but they do not indicate the potential corrosiveness of the steams carried. The same is true of the data collected from *main* return lines. Only in condensing equipment, or in branch return lines adjacent thereto, are the conditions of operation such that measurements of corrosion rates reflect the corrosiveness of the steam supply. Why this is true will become evident later.

While the data shown in Fig. 3 indicate for the most part that only a mild amount of corrosion occurred at such locations, and therefore predict that pipe failures are unlikely, they do not presage freedom from corrosion

Table 1—Analyses of Deposits Taken from Heating Equipment (Per Cent by Weight)

	A Radiator After 5 Yr Operation	Return Lines In			
	Bldg. 11	Bldg. 5	Bldg. S_1[a]	Bldg. S_2[a]	Bldg. S_{12}[a]
Moisture	.....	0.51	0.01	0.46	0.50
Silica	8.9	0.59	1.90	3.62	0.28
Iron oxides (and alumina)	86.5	98.10	68.96	66.36	90.68
Ignition loss	4.9	.....	29.20	27.92	8.20
	100.3	99.20	100.07	98.36	99.66

[1]Progress report of NDHA corrosion committee (unpublished).

[a]No measurements of corrosion were made in these buildings.

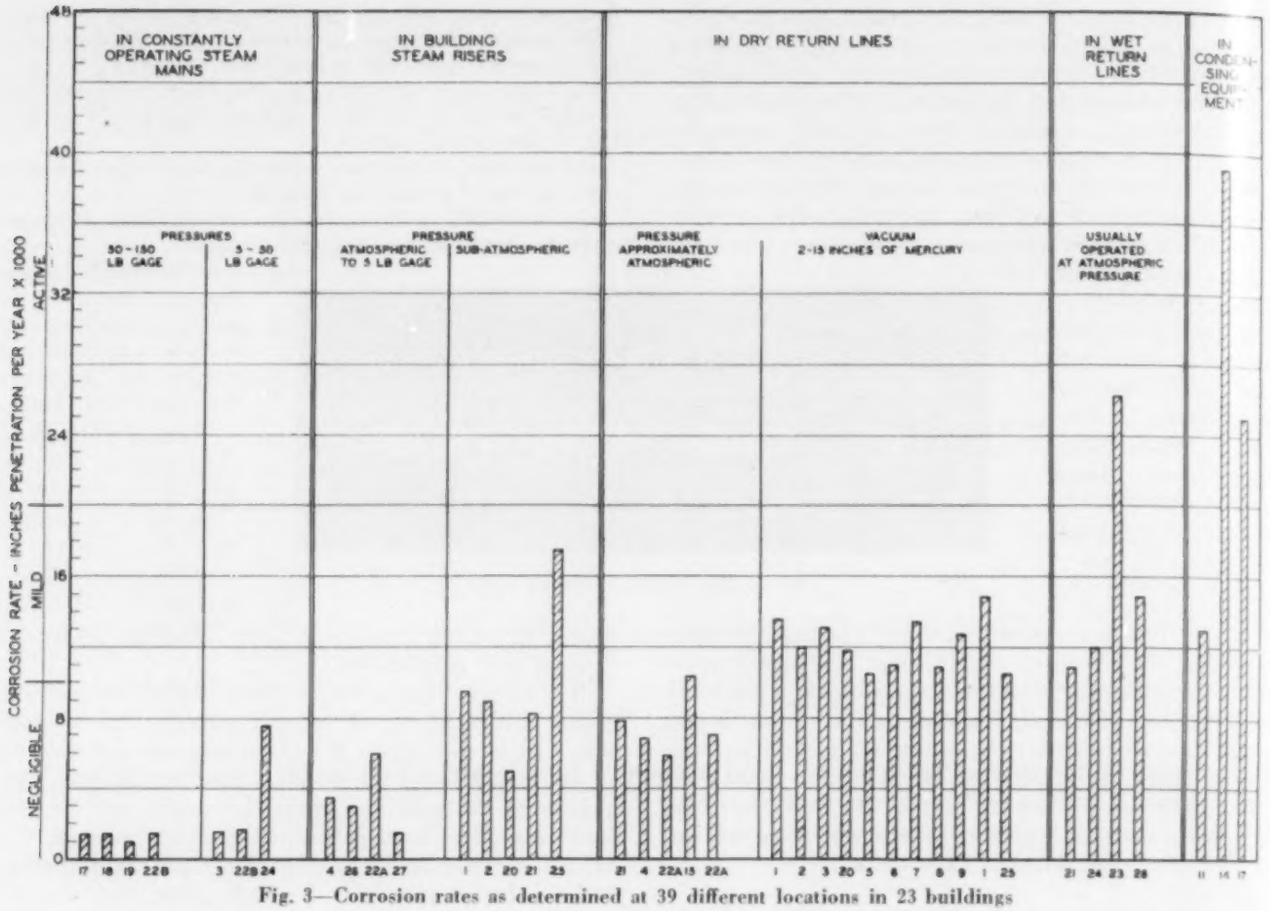


Fig. 3—Corrosion rates as determined at 39 different locations in 23 buildings

troubles. In every case some corrosion was detected. It is not unreasonable to suppose that if enough insoluble material so produced is caused to work along and accumulate at a restricted point plugging of the line will eventuate. That such deposits consist of materials of slight solubilities is attested by the analyses shown in Table 1.

The silica—particularly in buildings Nos. 11 and S_2—is believed to have originated largely from core sand. In building No. 5 failed pipes have been reported. At the other locations, especially in building S_{12}, the accumulations were so numerous that their removal presented a vexing problem.

[To be continued]

Control of Atmospheric Impurities

An answer to the question, "How clean is the air where your men work"? may be found through procedures given in a bulletin issued by Air Hygiene Foundation, 4400 Fifth Ave., Pittsburgh, Pa., to its 250 member companies and associations. Air impurities cause much of the illness among heavy industry's 15 million workers.

The bulletin, "Routine Sampling for Control of Atmospheric Impurities," was prepared by the foundation's preventive engineering committee, headed by Philip Drinker of Harvard. It lists proved methods of collection, determination and apparatus for checking 45 substances found in industrial atmospheres, such as ammonia, arsenic dust, benzene, chromic acid, carbon dioxide, hydrocarbons, lead dust or fume, mercury, methanol, sulphur dioxide, etc.

Two steps are involved in checking the air of a workplace, the report explains, "first, measurement of air currents, and, second, estimation of impurities." The resulting data can then be applied to the design of control equipment or to checking the results brought about by such equipment. If control equipment has been installed, samples should be taken periodically for check-ups.

The samples are about the only proof that the employer can bring forth that he has taken proper precautions to prevent a disease, such as silicosis or a chronic poisoning from benzol. The mere statement that he or anyone else thought the atmosphere sufficiently clean is not enough. Occasional photographs of various plant and mine operations are often of greater practical usefulness than are records of dust counts or of gas or vapor concentrations.

Air Conditioning in the Tropics

The American Geographical Society, Broadway at 156th St., New York, N. Y., has recently issued a 32 page booklet "Tropical Climatology and Physiology in Relation to the Acclimatization of White Settlers" by Robert G. Stone of the Blue Hill observatory of Harvard University. It comprises appendices and footnotes reprinted from a society publication entitled "White Settlers in the Tropics."

The subjects included in the booklet are basal metabolism; body heat production; body temperature regulation; effects of heat loss; sweating; tropical effects on blood, viscera and body tone; comfort zones and acclimatization; a note on the cooling power (which is defined as the net rate at which heat can be taken up from any body by its environment); etc. A number of references on physiology and acclimatization in the tropics, and air conditioning in the tropics are given.

How Can We Lighten the Load of Air Conditioning Selling Costs?

By Lester T. Avery*

GOOD morning . . . Is this the Whozis Company? Could you send a man over to give us a price on air conditioning? This is the Ninety-Ninth National Bank building, and we are considering air conditioning for several floors, or possibly the entire building. . . . No, we have no plans or specifications, and we would like each bidder to make up his own recommendation and proposal. . . . Yes, our building superintendent may have some old plans that you can look over, but there have been a number of changes made so you will really have to check the building itself. . . . No, we haven't determined any of the load factors; you certainly ought to know enough to do that, that's your business. . . ."

Oh, boy! Here's a good prospect!!

"Please work up the best proposition you would recommend; that is, the best system with the lowest first cost and operating cost. Something that would be just right for this building. Please separate your figures so we will know what it will cost to do the basement and first floor only, third floor, seventh floor, and eighth floor, and another proposition covering the entire building. . . . Yes, we would like all these prices separate with complete specifications and some kind of drawings which will show how you plan to install the equipment. You understand we don't want to give up any usable floor space. . . . Oh, yes, we are having all the better companies figure on this work as most of them are customers of the bank. . . Please include everything—that is, a turn-key job."

Hmm. That doesn't sound so good.

The above is exaggeration only in that it condenses into a one-sided conversation the pertinent facts developed during several phone calls and probably one or two interviews. Most work of this kind starts out innocently enough as a simple preliminary "guesstimate," and proper handling develops the customer's interest to the point where he really wants and needs more detailed information and competitive quotations. What would you say if you were on the receiving end of the above inquiry?

Growth of the Consultant

In the early 1900's most of us used the horse and buggy for personal transportation. We had horse and buggy ideas on construction work, and there is nothing really wrong with horse and buggy thinking, except that it does not fit today's conditions. I recall hitching up the old mare to drive my father over to see "how the house is getting along." He would buy vacant property, generally in acre or more sections, and gradually build this up with houses, either selling or renting; there always seemed to be at least one house under construction all the time. In those days the cost of building was very low, labor and materials were cheap, and a lot of work was done for a day's pay. It wasn't so important to squeeze the last dollar out of the contract cost, as the general real estate market was rising, and changes could be made as the house was built without running into severe losses.

The building contractor was really the carpenter-contractor. He would look at the lot and discuss informally the general type and size of the house, and give my father an estimate of the cost, which in those days ran around $3000 to $4000 for a seven or eight room house. A meeting of the minds took place and the builder would make some rough sketches which the family would check over just to be sure that there were closets and doors and other things that suited somebody's preference. We all felt qualified as building experts. The carpenter-contractor-builder handled all details; that is, he took care of the excavating, masonry, plumbing, heating, and electrical work. On the whole, the relationship was very satisfactory.

Those building contracts were more or less informal between the builder and the owner, and there was very little misunderstanding or dispute. As costs increased up to the time of the war, it became more important to be sure of details. We found minor changes that used to cost only a few dollars would run into substantial sums. The services of a professional architect became necessary, particularly in that he could design a building more economically, write a clear specification, and coördinate the dif-

If the selling outlet of an air conditioning equipment manufacturer—whether it be branch office or distributor-contractor — figures 10 sizable projects, five of which do not go ahead, and the other five are divided in the letting to the field, he may have one sale to carry the entire overhead burden of engineering-estimating on the 10 jobs. The problem of how to reduce this expense without lessening promotion of air conditioning sales is one that is discussed formally and informally wherever members of the industry meet. . . . Mr. Avery believes that an opportunity exists for consulting engineers and engineers associated with architects to take a more important part in promotion of air conditioning for both new and existing buildings, with benefit to all concerned—including the owner

*President, Avery Engrg. Co.

ferent branches of the work. As the size and cost of building increased the architect found need for the professional engineer who could handle the mechanical trades such as plumbing, heating, and electrical work. With the building boom of the 1920's there was a tremendous growth in architectural and engineering firms, and this talent is still available, waiting for the building industry to come to life.

An Opportunity

Is it pertinent to suggest promotion of air conditioning projects offers opportunity to engineers associated with architects and to consulting engineers? So long as the promotional work is done by the manufacturers and sales representatives, the selling cost is high and the estimating cost is high. In the bank building example mentioned above, six or more companies could spend from $500 to $1000 each in getting together the information that the owner requested. If the job goes ahead, only one company will secure the business and thus make what is called "overhead." If there are six bidders and six jobs and each bidder gets one, the overhead for engineering and designing must include this expense on the five jobs the bidder lost as well as the one he secured. The additional problem exists that many jobs do not proceed to an award of the contract, and the air conditioning sales offices have many files of complete plans and specifications on proposed work that never went ahead. In some cases where aggressive selling tactics have been followed there may be five jobs figured for one to go ahead. Fatalities in the dealer-contractor-distributor set-up are due partly to this tremendous overhead in relation to the business secured.

Is there any answer to this particular problem, and does the answer include the solution to high costs? For example, if the selling outlet of the air conditioning equipment manufacturer, whether it be branch office or distributor-contractor, figures conscientiously 10 sizable projects, five of which do not go ahead, and the other five are divided in the letting to the field, he may have one sale to carry the entire overhead burden of engineering-estimating on 10 jobs. Readers who may be in this business can check against their own records and will probably say that is a conservative rather than a liberal ratio.

Who pays the cost of estimating and designing? Obviously, the customer pays if he deals with a contractor who is staying in business. The manufacturer pays if he sells to contractors who disregard this expense and go out of business; the customer also pays indirectly for this loss. You have frequently heard it said, "We will put in air conditioning when costs come down," and the remark is most often made by the purchaser who expects the selling end of the industry to carry this extraordinary sales expense. Several solutions have been suggested, but there is no one of them that suits all interests.

The manufacturer has carried much of the burden of sales promotion. He could refuse to do any further "free" engineering and he could discourage his sales branches or dealers from so doing. Then what becomes of sales volume? Without some adequate substitute it would be suicidal for the manufacturer or his representative to take an arbitrary position and say, "Go hire yourself an engineer." The present volume of work is due largely to promotional work done by those directly interested in the sale of the refrigerating equipment and air units used in the air conditioning system. Yet these people would welcome a sensible solution.

The independent consulting engineer could play an important part if he would insist that *every* new building for public use included air conditioning and incorporate this in his plans. The same is true of the architect. But what about the vast market of present buildings, hotels, hospitals, stores, offices and commercial buildings that are the present fertile field? Can the individual consulting engineer profitably undertake the sale of his services on this type of work?

How Can Consultant Sell His Services?

It might seem strange for the consulting engineer to start out to canvass possible work, make a sales presentation of the idea, and then sell his services to the owner on a fee basis. On small work the fee would hardly pay the cost. Would it be better if he had an arrangement with the contractors whereby he could sell the survey with specifications to the bidders, either at a lump sum price divided equally between the number of bidders, or on a percentage basis, say one per cent to each bidder, with a maximum set of perhaps five bidders? The sixth bidder would be discouraged from figuring on the basis that five bids on one job are enough. If this were handled through a consulting engineers' central bureau there should be little cause for complaints of unfair handling, and the bureau could employ a paid salesman-secretary who would solicit the work and refer it to the individual members in turn to design, and then in consultation with the owner select the five bidders. The contractors in turn would refer inquiries (such as the one described in our first paragraph) to this engineering secretary. A bureau of this kind would be required in all major cities.

If the above suggestion does not seem practical, would it be better for the contractors to set up a central survey bureau, employ a group of designing engineers, and refer all prospects to this bureau for survey and preliminary specification? Each contractor could use the services of the bureau on a fixed fee or percentage basis, and if other contractors solicited the same job they would have the right to the survey by paying their share of the cost. In this way, if only two or three figured, the engineering cost would be divided in two or three parts. Also, if there were a clearing house, the bidder that found there were already five or more bidders on the job would probably stay out unless he felt there was some particular reason for his coming in.

With both of the above plans, the ingenuity of the contractor and the use of his own special equipment could be fully utilized and there would be variations in bidding and true competition. It would help in the solution of the problem of why there is sometimes 100 per cent variation in capacity and price from different bidders. The difference is due to engineering analysis.

Function of Consulting Engineer

No reliable contractor objects to competitive bidding if it is honestly competitive. It costs very little to figure from a complete plan and specification. The difficulty arises when we hear the remark, "You fellows can't agree on what is required." If plans and specifications are not complete in detail, various bidders may analyze the problem in different ways—with the result that widely varying quotations are submitted. This results in loss of confidence on the part of the purchaser. It sometimes happens where an architect takes bids on air conditioning without setting up a definite engineering specification. It is even more common when the owner takes bids direct. The criticism is unfair, as air conditioning is an art as well as a science, and every competent engineer who analyzes the problem will arrive at a solution based on his own training and experience. What does the owner want, and will he pay for it when he says he wants it? Should the job be figured for all-year temperature control, zoned to take care of sun and wind? Is accurate humidity control desired, winter and summer? Is the air to be cleaned, and if so, what is clean air? What is noise, and how much of it can be tolerated in this particular installation? What about drafts, and whose opinion is to be considered? How much outside air is required for good ventilation? How and where can the apparatus be placed, and will it mar the general appearance of some of the finished rooms or can it be concealed?

The consulting engineer settles these questions, presumably in agreement with the owner, and he makes the compromises that may be necessary in order to apply air conditioning to an existing building. He also relates the equipment to first cost and operating cost so that the first class building has the best possible air conditioning system, and the simpler commercial work has possibly only comfort cooling. Either of the plans mentioned above would operate in this way. The wide variation in quotations comes when two or more conscientious designing engineers have to settle those questions with an owner of limited experience. Then the bids are not competitive and in many cases the installation results in disappointment. For example, an owner will recall all the pertinent points raised in discussion by various bidders, each emphasizing his own particular offering, but when he comes to buy, cost becomes of exaggerated importance and his purchase may not include a number of the features he has discussed and thought he was getting. Particularly is this true in the control and ventilation details where zoning and economizing control, humidity control distinct from temperature control, etc., all sound very desirable and simple but can be omitted at considerable saving in cost.

It does not help to employ the professional engineer to check bids after they have been submitted—that is, it doesn't help from the standpoint of cost of figuring, although the consultant may help the owner in guiding to an intelligent selection. This is "locking the barn after the horse is stolen," as every bidder has by that time incurred his estimating and engineering expense, possibly with two or three revisions. When the consulting engineer comes in late he has two separate and distinct problems. One is to save enough in the letting of the job to justify his fee, and the second is to put his own personality into the design so that his client gets the benefit of the consultant's experience—which may require all bidders to re-figure. If it should so happen that the consulting engineer sets up more rigid requirements than any one of the bidders, the new quotations come in higher than the original ones, and the owner says, "Well, that's what happens when you get a consulting engineer. Instead of saving money he actually costs you money." This again is not a fair criticism, but the industry suffers because of it.

The consulting engineer or the architectural firm with its own engineering staff has the greatest opportunity in undertaking the initial promotion and selling its service along with the general idea of using air conditioning. This will relieve the sales and estimating end of the business of considerable cost, which will be reflected in lower cost for work quoted. This, in its turn, will widen the market for air conditioning and a larger volume of business will inevitably come with lower costs. Material costs are rigid, and in the larger cities labor costs are fixed, with the trend upward. How can you satisfy the need for lower cost construction work, particularly in the field of air conditioning, unless you eliminate some of these unnecessary costs? Free engineering, free designing, and free estimating are not free, and the purchaser pays for this expense on all the work that is being figured.

A Specialized Problem

There is considerable responsibility in air conditioning work that indicates the need for continuity. The maintenance of refrigeration or dehumidifying equipment at full efficiency, the adjustment of controls, and the continued economical operation require the services of the trained engineer, generally in the employ of the air conditioning contractor. These men are available to coöperate with architects and consulting engineers in the planning of the work, and in turn take the responsibility to maintain the equipment and live with the installation during the next 20 or 40 years, whatever the life of the building may be.

In this sense, air conditioning is a little more specialized than plain heating and ventilating. It was relatively simple to plan heating equipment somewhat oversize as the only criticism would come from insufficient heat. Overheating was a matter of operation on the part of the owner after the engineer and contractor were finished. Ventilation likewise was simply supplying air or exhausting it—or both—with rarely any effort made to gain the economy of recirculation. It has been said that air conditioning is "nothing but glorified hot air heating." Where is the glory and who gets it? Is it not true that air conditioning is a result of the perfection and development of heating, ventilating and cooling, and the owner who buys air conditioning expects to get the results he has *not* been getting with ordinary heating and ventilating, and "summer wind" cooling. He really wants *clean* air with temperature and humidity *control*, freedom from drafts, sufficiently quiet to please.

Yet air conditioning is not simply the engineering

problem of temperature and humidity control. Air is supplied to people to breathe and live in and there is a wide difference in the needs and preferences of people as to what is just right. Dr. James Roeder Bell points out that while 90 per cent of the people are normal and respond favorably to what is called normal comfort air conditioning, there is a fringe of 5 per cent who are feeble or aged or who have very low metabolism who require higher temperatures and less air motion for comfort or even health, and there is another fringe of 5 per cent who have high blood pressure, intense nervous reactions, and require lower temperatures and more air motion than the normal person could tolerate. Instead of trying to make the air conditioning system fit this fringe of the public it is better to design for the average *normal* person. The individual who is not comfortable in the combinations of temperature, humidity and air motion that have been determined as normally satisfactory is in reality a medical case rather than an air conditioning problem.

Everyone in the industry—architect, engineer, contractor, and manufacturer—can help in promoting the wider use of comfort air conditioning by a mutual recognition of this physiological fact. People are different, and a *perfect* air conditioning system may be criticized by a few *abnormal* persons.

In Conclusion

Some architects may look on air conditioning as a nuisance. They must make beams where they want no beams, and plan for grilles where they prefer a plain wall. Other provisions must be made in construction to permit the installation. While most of the consulting engineers or engineers connected with architectural firms encourage the use of air conditioning, there are still a few who are apathetic because they do not understand air conditioning and are confused by the variety of opinions expressed by supposedly qualified experts. The manufacturers and contractors are struggling with high costs and limited sales at profitless prices.

The owner of the modern store, or any building serving the public or a group of employees, knows full well that it is obsolete without air conditioning. He needs air conditioning for business, for comfort, and for health. He is not sure where or how to get it. The manufacturer and the contractor-engineer have been carrying most of the promotion burden. How would you answer the inquiry presented in the first paragraph? Would you spend *your* time and money to go into this job along with five to 10 of your competitors? Would you recommend an architect and consulting engineer be selected? Would you not use a central survey bureau maintained by the consulting engineers as a group or the contractors as a group? Surely there must be some way by which these groups can coöperate to reduce the cost of the work, increase more rapidly its acceptance, insure the financial stability of the industry, and increase the use of air conditioning for better living.

Texas A & M Holds Air Conditioning Short Course

NEARLY a hundred persons attended the first annual air conditioning short course held at the Agricultural and Mechanical College of Texas, College Station, Tex., August 17-19. The meeting was sponsored by the school of engineering and was held under the auspices of the mechanical engineering department. At the banquet held on Friday evening, John Howatt, chief engineer of the Chicago board of education, past president of the American Society of Heating and Ventilating Engineers, and a member of HPAC's board of consulting and contributing editors, spoke on the duties, responsibilities, and obligations of engineers designing, selling, installing and operating air conditioning equipment.

The address of welcome to the short course was given by Gibb Gilchrist, dean of engineering, who expressed his pleasure at the large attendance and told the group that the two important features of engineering education at the present time are air conditioning and aeronautics, and that the college will use its facilities in research, education, and extension to assist in the development of these important phases of engineering. Dr. F. E. Giesecke, director of the engineering experiment station of the college, presented a paper on research as related to air conditioning, in which he reviewed briefly the various coöperative research projects of the American Society of Heating and Ventilating Engineers. He called particular attention to research projects now under way at the college, principally research relating to the construction of fans; the use of fans in removing heat from buildings by means of air currents in connection with a college auditorium seating about 2000 people, in which the sensible heat produced by the occupants can be removed with an increase in temperature of 10 F of the air in flowing through the building; the use of gravity ventilation in coöperative dormitories recently constructed, covered with corrugated galvanized tin roofs, in which indoor and attic temperatures were lower than in a corresponding building covered with wooden shingles; the time lag in the flow of heat through a brick wall; the use of water jet pumps utilizing the available energy in the flow of water instead of motor driven circulating pumps in connection with central hot water heating systems; and the use of heating pipes in panel warming systems as reinforcement for concrete slabs.

The program also included discussions on air conditioning for human comfort, by Prof. H. E. Degler, University of Texas; heat gain and loss calculations, by A. M. Chase, York Ice Machinery Corp.; pneumatic controls, by J. D. Poythress, Minneapolis-Honeywell Regulator Co.; evaporative cooling for automobiles, by J. W. Miller, Texas Electric Service; air conditioning equipment, by Hal Gibson, General Electric Co.; air distribution, by Herbert Kunen, Anemostat Corp. of America; air conditioning controls, by H. E. Shugars, Barber-Colman Co.; chemical humidity control, by W. E. Stark, Bryant Heater Co.; side wall distribution of air, by Mr. Shugars; and trouble shooting, by A. J. Rummel, San Antonio Public Service Co. Proceedings of the short course are to be published and sent to all those who paid the registration fee of $1.00.

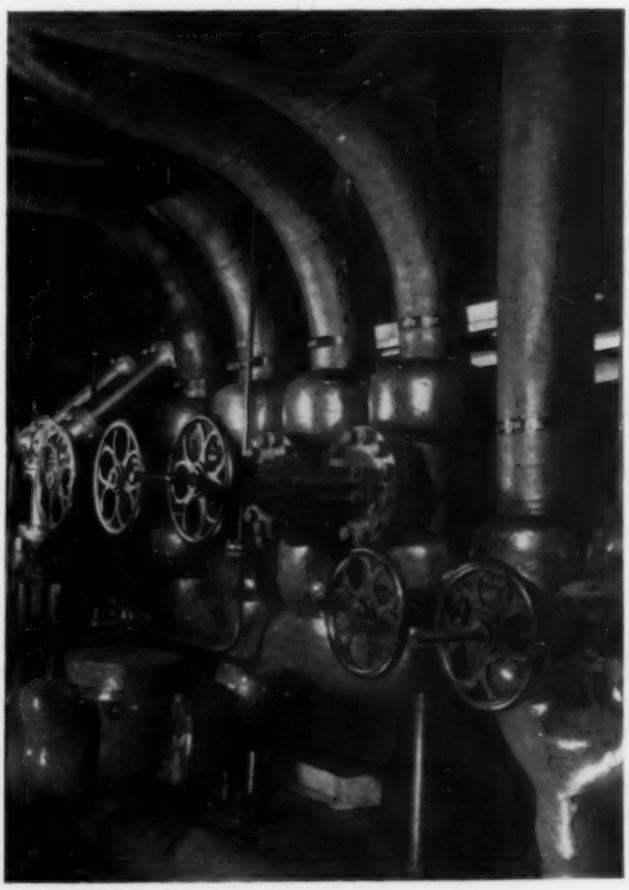
Establishing Design Pressures for High Pressure Boiler Feed Piping

By H. N. Boetcher*

GENERALLY speaking, the design of boiler feed piping is not one of those factors in power piping which greatly affect the total cost of a system. This may be one reason why it has been handled so long as an adjunct to the steam piping. The advent of steam pressures in the 450 lb class some 10 to 15 years ago first forced more careful consideration, as feed pressures often were in the range of 550 to 600 lb per sq in. The extensive use of steam pressures of 800 to 900 and 1250 to 1450 lb per sq in. in recent years, together with the increase in steam temperatures to 900 F and higher, has changed the situation radically. The use of different materials for steam and feed water piping emphasized the absurdity of using one basis of design. In addition, temperature (as well as pressure) has become more fully recognized as a factor of importance. As a result it has been realized rather generally now that high pressure feed piping should be designed on the basis of the temperatures and pressures actually encountered. The temperature, of course, is a rather definite figure and can be considered on the basis of the temperature correction curve for allowable stresses given in the ASA code for pressure piping. The pressure is less easily determined and depends on conditions as well as the importance which the designer might attach to various factors. A brief analysis of conditions as made in connection with the design of a high pressure extension now under construction may therefore be of more or less general interest.

The carbon steel pipe normally used in boiler feed systems is relatively inexpensive and its cost is, therefore, not a major factor. The cost of valves is somewhat more important. The increase of rigidity and of expansion and contraction forces resulting from increased wall thicknesses, is, however, of considerable importance. The actual selection of thicknesses is bound to be influenced, therefore, by plant conditions, with a tendency toward a "safer" design with heavier walls in plants with opportunities for liberal design and toward a close calculation where space limitations force a crowding of the piping. As it is probable that thicknesses based on close design will ultimately be adopted generally when proved safe, the determination of the safe basis of such a design is important.

The advent of higher steam pressures and temperatures in power plants has forced more careful consideration of the design of boiler feed piping, and it is generally realized now that such piping should be designed on the basis of the temperatures and pressures actually encountered. Mr. Boetcher discusses here the factors influencing the pressures in boiler feed piping under various conditions, and calls attention to the margin of safety between hydrostatic test pressures and potential maximum operating pressures which should be specified in design.

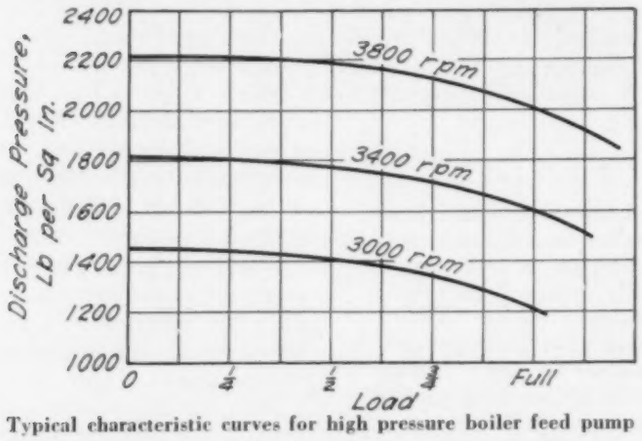


Both the boiler code and the code for pressure piping link boiler feed piping to steam pressure ratings. Neither code fixes expressly the method of determining the working pressures. The boiler code, however, refers to "maximum allowable working pressures" which are subject to fairly accurate calculation as the piping covered by this code is located beyond the regulating valve. A reference to temperature in the boiler code is of little practical importance in feed piping design as long as the lowest temperature for which stresses are set is 700 F. The code for pressure piping ties ranges of boiler feed pressures to the steam pressure classes, without reference to temperatures and without any statement as to the method of determining the design pressures. The pressure ranges of boiler feed piping assigned for the 900 and 1500 lb steam classes are so broad that piping may be appreciably over- or under-dimensioned.

*Assistant to Superintendent of Power Production Stations, Consolidated Gas Electric Light and Power Co. of Baltimore.

Factors influencing the pressures in the boiler feed piping are possible shocks, characteristics of boiler feed pumps, and characteristics of the pump drive. Another factor to be considered is the extent to which the margin between hydrostatic test pressures and normal maximum pressures could be called upon to take care of unusual and infrequent increases in pressure.

Shock pressures are the results of rapid closing of valves, especially of regulating valves. These forces are indicated occasionally by sudden movements of the pipe. Obviously, the rate of closing of regulating valves is a controlling factor and should be kept as low as is compatible with reliable operation. Analysis of the conditions indicates, however, that the rates of closing normally to be expected do not cause dangerous increases in pressure. Movements of the pipe, as observed, may be misleading regarding the seriousness of the shocks as affecting internal pressures though they may indicate a necessity for reviewing the anchoring and supporting set-up of the piping. Modern methods of pipe support often give a surprisingly flexible system in which considerable movement may be caused by comparatively small forces. In one recent case, an 18 in. header designed for 300 lb per sq in. pressure was moved by hand. Shocks, in such cases, are of concern with respect to supports rather than determination of wall thickness.



Typical characteristic curves for high pressure boiler feed pump

The pump drive is a factor of paramount importance. With variable speed drive, the boiler feed system pressure is controlled by pump speed, on the basis of pressure differential at the regulating valve or some similar factor. With decreasing load, therefore, the discharge pressure of the pump is lowered. This condition is reversed with constant speed drive as the pressure follows the head capacity curve of the pump. Since this curve has to have a "drooping" characteristic for satisfactory operation of pumps in parallel, a lowering of the load results in increased pressures.

It appears, therefore, that with variable speed drive maximum pressures occur at maximum flow through the pump. If two or more pumps serve a boiler or a jointly operated bank of boilers, the maximum pressure in the piping would correspond to the discharge pressure of any pump at maximum speed and output. With one pump serving a boiler or bank of boilers, the maximum pressure would be obtained with maximum evaporation of the boilers, all safety valves blowing, which would be less than maximum pump capacity. The selection of the design pressure for the piping should be governed accordingly by the plant layout. In addition, to meet the emergency condition that the speed regulating mechanism might stick at maximum speed and that the pressure then might increase to the shut-off pressure at this speed, we feel that the stresses in the piping should be at a safe level at this condition. This level has been set tentatively by us at not more than one-half of the specified minimum yield point of the pipe steel.

With constant speed pump drive, operating conditions again influence the predominant pressure in the piping as operation of the pumps at high load would tend to keep the pressure relatively low. On the other hand, variable load results in pressures which frequently approach shut-off pressures, as the characteristic curves (especially of high pressure pumps) are comparatively flat even up to nearly half load. As every installation sooner or later is operated at less than full load, the shut-off pressure of the pumps appears to be the only safe basis of design. This is true even when the normally used pumps have variable speed drive and a constant speed emergency pump is installed, as the latter pump may and will occasionally supply the feed water demand.

A factor which should receive more general attention is the margin in safety offered by the differences between hydrostatic test pressures and potential maximum operating pressures. This margin, in many cases, is quite small and sometimes is even negative. Present requirements of the code for pressure piping require hydrostatic tests at twice the normal service pressure, but not more than 1½ times the primary steam service pressure rating of the valves included. In a specific job now being designed, the test pressure would be 2250 lb per sq in., with a contract pump shut-off pressure at maximum speed only about 100 lb per sq in. less. This margin is too low and actually may become negative if the pump characteristics vary but slightly from those expected. Such cases have been reported by others. It appears that power piping practice could well afford to follow the lead of the oil industry which specifies hydrostatic test pressures of twice the primary rating for gate valves (API Standard No. 600A-37T).

Manufacture of Brass and Copper Pipe

Brass and copper pipe and tube are manufactured by one of several different processes, as follows:

1) Sand or chill casting a tubular shell of the alloy, then removing, if necessary, the outside rough surface which may accompany such casting, and drawing and redrawing the resulting shell until the final tube size is reached.

2) Hot piercing, by which cylindrical bars of metal are cast, the outside machined and the bars then heated and by rotating and pushing them against a pointed cone a rough tubular shell is formed. This is then drawn to size as in the first method.

3) Cup drawing, by which process a flat bar of metal is cast and then rolled to about ½ in. thickness, the surface imperfections removed by machining, and discs blanked from this rolled metal. The discs are in turn cupped, the cupped end cut off and the tube then drawn to size as in the other methods.

4) The extrusion process. Solid billets made as in the second process are cut to the desired length. Then these are placed in a hydraulic press, the center punched out and the shell extruded between dies and a mandrel to form a shell which is then drawn to size.

Dioramas illustrate these processes at the Copper & Brass Exhibit at the New York Fair.

The Psychrometric Chart

Its Application and Theory

By William Goodman*

LINES of constant wet bulb temperature are not shown on Chart A because their location changes with every change in barometric pressure. If the chart is to be useful for a wide range of barometric pressures, the lines of constant wet bulb temperature cannot be shown. However, inasmuch as the lines of constant wet bulb temperature are straight lines on a chart of this type, they can easily be drawn by either one of two methods that will be given here. First, however, some of the theory necessary for an understanding of the subject will be presented.

Wet Bulb Temperature

Assume for purposes of discussion that water lies in an open shallow pan. The pan itself is set in an insulated horizontal duct through which air is flowing, and the air sweeps across the surface of the water. If the water is initially warm, it will at first cool rapidly and then more slowly until it finally reaches a constant temperature. Even though the water continues to evaporate, it will remain at this constant temperature. In the same way if the bulb of a shielded, accurate thermometer is covered with a wet wick, and placed in a stream of swiftly flowing air, the water on the wick will soon cool to some constant temperature at which the water will continue to evaporate. Inasmuch as the bulb of the thermometer is covered with the wet wick, the thermometer itself will indicate the temperature at which the water is evaporating from the wick. Because the temperature of the evaporating water is measured in this way, and also because the constant temperature which the water will assume depends upon the condition of the air flowing past the wick, the temperature at which the water evaporates is commonly called the *wet bulb temperature* of the air.

In the opening paragraph of this section, it is stated that lines of constant wet bulb temperature can easily be drawn on the psychrometric chart. Actually, for reasons that will be given later, lines of constant wet bulb temperature cannot be drawn on any psychrometric chart. A substitute for these lines of constant wet bulb temperature is commonly used on all psychrometric charts. Instead of lines of constant wet bulb temperature, practically all charts show lines of constant *temperature of adiabatic saturation*, which are defined in the following paragraphs. The lines of constant temperature of adiabatic saturation that are so drawn are commonly called lines of constant wet bulb temperature, though strictly speaking the two temperatures are not equal to each other. However, for water and air, the difference between the wet bulb temperature and the temperature of adiabatic saturation is very small.

The slope of any ratio line drawn through a point such as *1* of Fig. 8 depends upon the initial temperature of the water that is sprayed into the air. As the initial temperature of the water supply is increased, the slope of the ratio line will also increase. The ratio line for water at 32 deg intersects the saturation curve at a dry bulb temperature above 32 deg. On the other hand, the ratio line for water at 212 deg intersects the saturation curve at a dry bulb temperature below 212 deg. In neither case is the temperature through the intersection point on the saturation curve equal to the temperature of the water for which the ratio line is drawn. Clearly, in between these two extremes there must be one ratio line that will intersect the saturation curve at a temperature that is equal to the temperature of the water for which the ratio line is drawn. This is illustrated in Fig. 9 where the slope of the ratio line *1-2* corresponds to water whose initial temperature is equal to the dry bulb temperature line through point *2*.

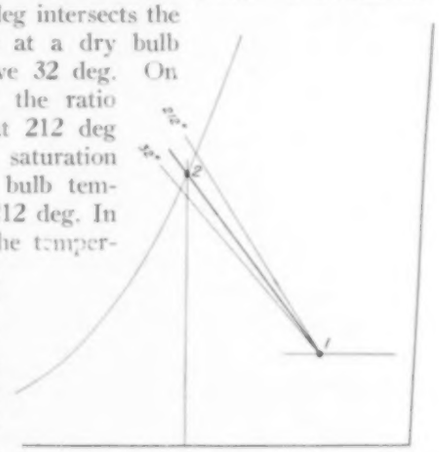


Fig. 9—Diagram illustrating temperature of adiabatic saturation. The slope of the line *1-2* corresponds to water whose temperature is equal to the dry bulb temperature line through point *2*. In this illustration the angle between the lines has been exaggerated for clarity

For any point such as *1* of Fig. 9 there is only one ratio line that will intersect the saturation curve at a

In addition to the elementary uses of the psychrometric chart—such as showing the relationships between dry bulb, wet bulb, and dew point temperatures, and relative humidity—it provides a simple and enlightening means of analyzing many complex problems in comfort and process air conditioning, including drying. . . . New psychrometric charts covering the low, middle and high temperature ranges, with saturation curves for various barometric pressures, are presented here. The general theory of the psychrometric chart is discussed in detail, and the methods of solving problems by the use of charts are described. . . . These new charts facilitate solution of most problems and lead to clearer understanding of the fundamentals

*The Trane Co. Member of Board of Consulting and Contributing Editors.

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dry bulb temperature equal to the temperature of the water for which the ratio line is drawn. Another way of clearly illustrating this point is shown in Fig. 10, where *2* represents a point on the saturation curve. The ratio lines corresponding to water at 32 deg and 212 deg are drawn through this point. The slope of the ratio line *1-2* corresponds to the enthalpy of water that is at a temperature equal to the dry bulb temperature line through point *2*. Manifestly then, at any one point on the saturation curve there is only one ratio line whose slope corresponds to the enthalpy of water that is at a temperature equal to the temperature of the saturated air. This one ratio line is called a *line of constant temperature of adiabatic saturation.*

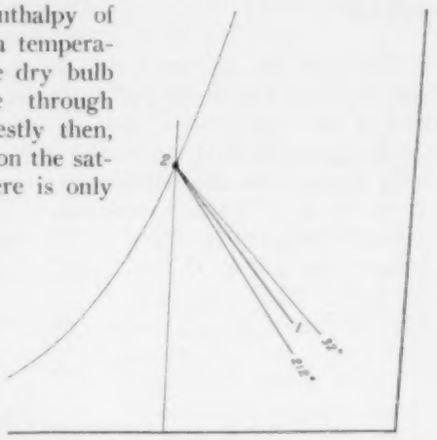


Fig. 10—Line of constant temperature of adiabatic saturation drawn through a point on the saturation curve. The slope of the line *1-2* corresponds to water whose temperature is equal to the dry bulb temperature line through point 2. In this illustration the angle between the various lines has been exaggerated for clarity

The temperature of adiabatic saturation owes its importance to the fact that the wet bulb temperature of the air is very nearly equal to the temperature of adiabatic saturation if the air velocity is sufficiently high. Arnold[9] has shown both by theory and experiment that the difference between the two is negligibly small for water and air although this is not the case for liquids and gases other than water and air. This has also been demonstrated by Carrier and Mackey[10], and by Dropkin.[11] For most purposes, the wet bulb temperature of the air may be said to be equal to the temperature of adiabatic saturation.

The wet bulb temperature—that is the temperature which the evaporating water will assume—depends upon the velocity of the air passing over the wet bulb of the thermometer. For this reason, more than one wet bulb line can be drawn through any given point on the psychrometric chart. On the other hand, the temperature of adiabatic saturation is independent of the air velocity. Consequently, only one line of adiabatic saturation temperature can be drawn through any given point on the psychrometric chart. Therefore, because the difference between the wet bulb temperature and the temperature of adiabatic saturation is negligibly small if the air velocity is sufficiently high, the lines representing the temperature of adiabatic saturation are commonly used on the psychrometric chart instead of lines of true wet bulb temperature. Inasmuch as the wet bulb temperature differs from the temperature of adiabatic saturation, the ratio line corresponding to the wet bulb temperature must be different from the ratio line corresponding to the temperature of adiabatic saturation. The two will, however, almost coincide because the difference in the two temperatures is negligibly small.

Because lines of constant temperature of adiabatic saturation are used instead of lines of constant wet bulb temperature, the phrase "wet bulb temperature" is frequently used where, strictly speaking, the phrase "temperature of adiabatic saturation" should be used. In order to conform to common usage, the term "wet bulb temperature" will frequently be used in the balance of this article instead of the correct, but more cumbersome, "temperature of adiabatic saturation." No confusion will result if the reader will bear in mind the fact that only lines of temperature of adiabatic saturation can be drawn on the psychrometric chart by means of the two simple methods that are presented here, and that these lines are used as approximations to the lines of wet bulb temperature.

As previously shown, when heat and water are added to air, the slope of the ratio line is determined by equation 3, which is†

$$\frac{h_2 - h_1}{w_2 - w_1} = \frac{Q}{W_o} + h_o$$

If the process is adiabatic, that is there is neither a gain nor loss of heat, then Q in the preceding equation is equal to zero. When this is the case, equation 3 reduces to equation 12, which is

$$\frac{h_2 - h_1}{w_2 - w_1} = h_o$$

In order to draw a line of constant temperature of adiabatic saturation through any point on the saturation curve, draw the one ratio line whose slope corresponds to the enthalpy of water that is at the same temperature as the temperature of the saturated air. Thus, in Fig. 10 the slope of the ratio line *1-2* corresponds to the enthalpy of water that is at the same temperature as the temperature of the saturated air in the condition represented by point *2*. Hence, equation 12 may be written.

$$\frac{h'_2 - h_1}{w'_2 - w_1} = h'_o \qquad [14]$$

where h'_2 and w'_2 are the enthalpy and absolute humidity, respectively, of saturated air in the condition represented by point *2* of Fig. 10, and h'_o is the enthalpy of water at the same temperature. Equation 14 may properly be called the fundamental thermodynamic equation defining a line of constant temperature of adiabatic saturation on the psychrometric chart. The equation for the temperature of adiabatic saturation used by Carrier[12] will be discussed later and shown to be a special case of a more general equation.

The following example illustrates one method of locating lines of constant temperature of adiabatic saturation—or lines of constant wet bulb temperature—on the psychrometric chart.

[9] J. H. Arnold, The Theory of the Psychrometer, *Physics*, Vol. 4, July, 1933, p. 255; September, 1933, p. 334.

[10] Carrier and Mackey, A Review of Existing Psychrometric Data in Relation to Practical Engineering Problems; *Transactions*, American Society of Mechanical Engineers, PRO-59-1, January, 1937, p. 33.

[11] D. Dropkin, The Deviation of Actual Wet Bulb Temperature from the Temperature of Adiabatic Saturation, *Bulletin* 23, Cornell University Engineering Experiment Station, July, 1936.

[12] W. H. Carrier, Rational Psychrometric Formulae, *Transactions*, American Society of Mechanical Engineers, 1911, p. 1005.

†See table of symbols, p. 358, June issue.

Example 7: On the psychrometric chart draw the lines for a wet bulb temperature of 70 deg for the following barometric pressures: (a) 24 in. (b) 31 in.

Solution:

a) Point *1* of Fig. 11 is located at the intersection of the line of 70 deg dry bulb temperature with the saturation curve for a barometric pressure of 24 in. For water at 70 deg, $h_s = 38$ Btu. Referring to Table 1 for $q = 38$, $\theta = -54°\text{-}25'$. Draw the ratio line *1-3* through point *1* of Fig. 11. This is the wet bulb temperature line for 70 deg at a barometric pressure of 24 in.

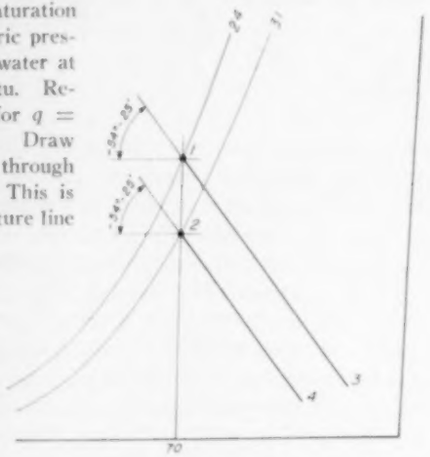


Fig. 11—Illustration for example 7. This diagram shows the method of using ratio angles for drawing lines of constant temperature of adiabatic saturation for air at various barometric pressures

b) Point *2* of Fig. 11 is located at the intersection of the line of 70 deg dry bulb temperature with the saturation curve for a barometric pressure of 31 in. Inasmuch as $q = 38$, the ratio angle is the same as before. Draw the ratio line *2-4* through point *2*. This is the wet bulb temperature line for 70 deg at a barometric pressure of 31 in.

As shown in another paper,[6] the lines of constant wet bulb temperature are very nearly parallel to the lines of enthalpy. In fact, the enthalpy along a line of constant wet bulb temperature increases only because of the heat of the liquid evaporated into the air. The latent heat required to evaporate this liquid into the air is provided by the cooling of the air itself. Accordingly, the only change in the enthalpy of the air-vapor mixture is the slight increase due to the heat of the small amount of liquid added to the air. This is discussed further in Appendix *3*.

When the wet bulb temperature of a point that is not located on the saturation curve is to be found, the two wet bulb temperature lines closest to the point should be drawn. Then the wet bulb temperature of the point itself can be read quite accurately. This is not quite the task it appears because the wet bulb temperature of any point on the chart can be approximated quite closely without even drawing the lines of constant wet bulb temperature. As previously stated, for most cases the lines of constant wet bulb temperature are almost parallel to the lines of constant enthalpy. Consequently, it is a relatively simple matter to determine which two wet bulb lines lie closest to the point whose wet bulb temperature is to be found. These two wet bulb temperature lines can be drawn on the chart and the temperature read accurately. For approximate calculations, the wet bulb temperature can be read with sufficient accuracy without drawing lines on the chart by using the lines of constant enthalpy as lines of wet bulb temperature.

[To be continued]

Health of Office Workers Aided by Air Conditioning

AIR conditioning accomplishes a 28 per cent reduction in those illnesses which cause an office employee to stay home from work, according to W. B. Henderson, executive vice president of the Air Conditioning Manufacturers' Association, in announcing the results of a survey of several large office buildings in various sections of the country which have kept statistical records of employee absenteeism.

In Cincinnati, the Procter and Gamble Co., which occupies five floors of an air conditioned building, reports that the loss of employees' services from colds and other respiratory ailments was cut from an average of 2.2 to 1.6 days per worker per year during the first 12 months of air conditioning, a decrease of 27 per cent.

In San Antonio, Texas, the United Services Automobile Association disclosed that absences due to illnesses were reduced from 7.5 to 5.5 days per worker per year with the installation of air conditioning, a decrease of 27 per cent.

In Philadelphia, the Philadelphia Electric Co. has been keeping absentee records for the past seven years, in order to compare the health level of employees in district offices equipped with air conditioning against those in offices without it. The number of days lost per 100 employees has been less in the air conditioned offices by the following percentages for each fiscal year: 1932-1933, 38 per cent; 1933-1934, 30; 1934-1935, 21; 1935-1936, 35; 1936-1937, 7; 1937-1938, 43; six year average, 29 per cent.

In the air conditioned Tribune Tower, Chicago, monthly records have been kept of the number of employees who were sick enough to stay home from work, and their total lost days. The improvement in health of Tribune employees is shown by the following reductions in total lost days per year from the average for the year before air conditioning was installed: 1934, 25 per cent; 1935, 46; 1936, 2; 1937, 25; 1938, 53; five year average, 30 per cent.

The epidemic of influenza which gripped the middle west in the winter of 1936 is readily apparent in the low figure of 2 per cent for that year. In commenting on this figure, however, J. P. Kreutzer, manager of the Tribune's employees' benefit plan department, said the staff suffered far less from the epidemic than Chicago firms which did not have air conditioning.

Medical authorities agree that the incidence of respiratory ailments in winter is minimized by air conditioning because a properly humidified atmosphere increases the resistance of the mucous membrane of the nose and throat to infection.

Mr. Henderson estimates that the extension of air conditioning to all workers throughout the country would effect savings of $600,000,000 per year in the present economic loss to industry due to employee illnesses.

Department Store Warehouse Heating Requirements Outlined by Specialist

By O. L. Merkt*

IN the past, the function of a department store warehouse was to provide for the storage of furniture and bulk merchandise until it could be placed on sale in the store and delivered to the customer. This is only one of the many activities carried on in the department store warehouse of today. In addition to warehousing, the delivery department with the necessary truck maintenance and repair shops and other service departments are situated in the building. These include furniture and cabinet shops, fur vault and fur repair department, carpet and drapery and other workrooms, facilities for laundry and dry cleaning, and many other service departments. Thus, the department store warehouse is really a service building.

The modern warehouse or service building is of fireproof construction, with reinforced concrete flat slab frame and roof and concrete or brick walls. The inside is generally unfinished except for plastered walls in toilets, offices and other special rooms. Insulation usually is not provided for the exposed walls, although insulating the roof is justified in many cases by its size and shape, local climatic conditions and the use of the top floor.

Inside Temperature—It has been found that a heating system designed to maintain different temperatures in different parts of a department store warehouse or service building does not give satisfactory results for many reasons. The most important is that a large percentage of the service building personnel, of which at least half are women, were formerly employed in the store building where temperatures are generally higher than necessary. The heating systems of old converted buildings (and of earlier warehouses designed for the purpose) where less than 70 F can be maintained have generally proved inadequate and the cause of complaints. As a result of these experiences, service building heating systems in the past 15 years have usually been designed to maintain 70 F.

Unit Heaters—In order to prevent over-heating in mild weather, unit heaters, each individually controlled by a mercury switch or similar thermostatic control, may be used. Unit heaters possess many advantages for this type of building. They can be placed on ceilings where they do not obstruct walls, aisles or storage racks; they can readily be moved or changed to suit any re-arrangement of the warehouse space; and few supply and return risers are required. Where a basement is not required, or where ground conditions make one undesirable, the heating returns from the first floor can be placed near the ceiling rather than in trenches.

Ventilating—A ventilating system usually is provided for removing exhaust fumes resulting from truck operations. Exhaust outlets are arranged under the platforms and connected through ducts to the exhaust fans. Unit heaters placed near the truck doorways provide the additional heat required to warm the air filtering into the driveway areas from outside. As the truck movements in the building are confined to short periods at the beginning and end of the day, the ventilating system usually is operated only during this time.

Piping Layout—The average department store warehouse and service building is from three to six stories in height. Therefore, the downfeed system is suitable for supplying steam to the heating system. Steam may be supplied at low pressure—say 5 lb—either directly from low pressure boilers or from high pressure boilers through a reducing valve, and carried by one or more main risers to the supply mains on the ceiling of the top floor. From these supply mains the various supply risers run down through the building. The unit heaters on the different floors are connected to the nearest riser.

The return risers begin at the unit heaters on the top floor and go down through the building, picking up the nearest unit heaters on each floor and connecting to the return mains on the ceiling of the basement or first floor.

Boilers—Except in the smaller buildings, the boiler equipment generally consists of water tube boilers suitable for 100 to 150 lb steam pressure. The heating load and the demand for live steam in many of the service departments are sufficiently large to justify this type of equipment. Usually two or more boilers are installed, depending on the space available and the load. When more than one boiler is installed, they are generally of the same size and capacity. Sometimes it is found advantageous to install two boilers of different capacities. In this event, the larger one is used for the entire steam load during the winter months, and the smaller one to furnish steam for the service departments during the remainder of the year. Usually this boiler has reserve capacity to carry a moderate heating load, if necessary, in addition to the service steam. During the winter

New warehouse and service building of Woodward & Lothrop, Washington, D. C.

*Vice president, Abbott, Merkt & Co., Inc., Industrial Engineers.

months when the larger one is in operation, the smaller boiler is a standby for emergencies.

Incinerator—Considerable rubbish, such as excelsior, wooden crates, paper, and so on, results from the unpacking of furniture, household goods and the like received at the warehouse, and an incinerator is installed for its disposal. In small warehouses where merchandise for storage may be received at intervals of a week or more, the rubbish resulting from the unpacking of this merchandise is sufficient only for the intermittent operation of the incinerator, but provides enough heat for hot service water. In larger warehouses where merchandise is received daily, there is sufficient fuel resulting from the unpacking of merchandise to operate the incinerator continuously. In this case the heat resulting from the combustion in the incinerator is used in one of the boilers.

Fuel—The selection of the kind of fuel for use in the boilers of a department store warehouse or service building is based on the same considerations that govern the selection of fuel for any other type of building. In a smaller plant, where the regulations permit the installation of a fully automatic oil burning system without constant attendance, oil fuel is usually found to be economical. In the larger buildings having boilers carrying 100 to 150 lb steam pressure, the plant will require constant attendance in any event, and the decision as to the kind of fuel to be used is a matter of comparative costs of operation of one kind of fuel as compared with another. Coal fired boilers are provided with automatic stokers suitable for the coal available.

Instruments—The extent to which indicating and recording devices are used on the boilers depends on the detail in which boiler operating data are desired. Usually the boiler plant in a modern warehouse is of sufficient size so that any slight increase or decrease in operating efficiency represents a neat sum in the cost of fuel either saved or wasted. Instruments that will assist the attendant in operating the boilers to the best efficiency are therefore a good investment always. At any rate, the boilers should be equipped with CO_2 and flue gas temperature recorders.

In locations where there are stringent smoke regulations, smoke indicators are provided to indicate to the fireroom attendant whether or not the boilers are operating in compliance with the regulations.

Storage is only one of the many activities carried on in the modern department store warehouse; it is really a service building, and may include the delivery department, truck maintenance and repair shops, furniture and cabinet shops, fur vault and fur repair department, carpet and drapery and other workrooms, laundry and dry cleaning facilities, and many other service departments. . . . The heating requirements of such structures are outlined here by the author, who has specialized in this type of work, with his recommendations for design details. The new warehouse of Woodward & Lothrop at Washington, D. C., serves as an example

New Woodward & Lothrop Warehouse

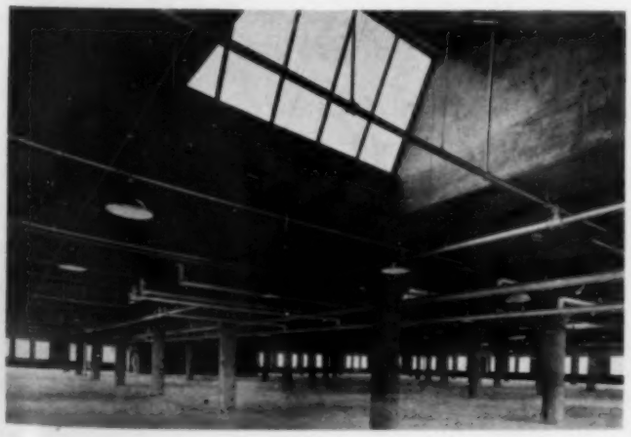
The recently completed warehouse and service building of Woodward & Lothrop in Washington, D. C., is a good example of a modern building of this type. This structure is six stories high, with a basement under the entire building. It has a frontage of 250 ft on the street, a depth of 259 ft, and is served by a railroad siding at the rear of the second floor. The basement contains a boiler room 50 ft wide by 125 ft long, extending in height from the basement floor to the ceiling of the first story. The remainder of the basement is used for storing house furnishings, wrapping and other allied activities.

The first or street floor contains the package delivery, a central wrap, offices and toilet and locker rooms. The furniture and bulk delivery and truck receiving are on the second floor. A ramp leads from the street level to the truck driveways on the second floor, providing direct access from the street. A first aid room, kitchen and dining rooms, and toilet and locker rooms are also situated on this floor, as well as the furniture finishing.

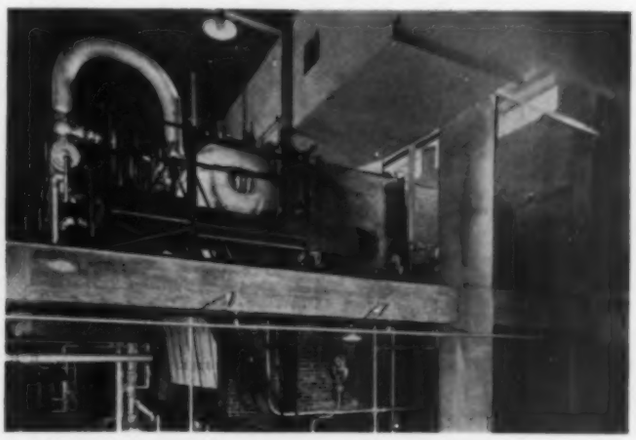
With the exception of a large fur vault and woodworking shop on the fifth floor, the entire third, fourth and remaining part of the fifth floors are used as warehouse space. The sixth floor is given over entirely to workrooms; these consist of a laundry and dry cleaning department, carpet and rug cleaning and carpet and drapery workrooms, upholstery shop and other smaller service departments.

With the exception of the offices, first aid room and

View of top floor, showing unit heaters and steam supply mains



Part of the boiler plant, showing the 550 hp boiler



A view of the stokers. The concrete ceiling above the firing space is the bottom of the coal bunker

toilet and locker rooms which are heated by cast iron direct radiation, the remainder of the building is heated by unit heaters. These are placed about 75 ft apart adjacent to the exposed walls, all pointing in one direction and discharging parallel to the wall. Additional units on the ceiling of the sixth story provide for the heat loss through the roof.

All the unit heaters are controlled individually through mercury switches and are connected to one central panel in the boiler room by individual floors. By this means, the unit heaters on an entire floor may be controlled directly from the boiler room; thus, at night or on Sundays and holidays, the unit heaters may be started or stopped, as weather conditions require, from one central point.

As the working area of the basement is entirely under ground, artificial ventilation has been provided. A fan discharges 20,000 cfm of warmed, filtered, outside air into one side of this area. One exhaust fan of 10,000 cfm capacity exhausts air from the opposite side of the basement and discharges it into the boiler room, where it passes out of the building through the boilers and incinerator, around the stack or through the windows. A second fan, also on the opposite side, exhausts 10,000 cfm from the basement and discharges it into the truck driveways on the first floor.

The exhaust fumes from the trucks in the first floor driveways are removed by two fans, each having a capacity of 10,000 cfm. This is discharged directly outdoors. The second floor truck driveways have ample unobstructed natural cross ventilation at the present time, and no mechanical system has been installed, although provision has been made so this can be done in the future if found desirable.

Steam is supplied to the heating system by the downfeed system, the distributing mains on the sixth floor receiving steam directly from the boiler room through one low pressure riser.

The boiler equipment consists of a 550 hp and a 200 hp cross drum straight tube boiler, built for a working steam pressure of 175 lb. The 550 hp boiler had been in use at the store, and was reconditioned and installed in the new building. The 200 hp boiler is a new boiler.

The two boilers are equipped with single retort unit type stokers, with individual fan and motor located at the stoker front. The 550 hp boiler is equipped with two stokers, designed to generate up to 700 hp, and the 200 hp boiler has a single unit designed to generate up to 400 hp continuously. The plant burns bituminous 1¼ in. nut, pea and slack screenings, with high operating efficiency.

The control features are thoroughly modern. Each stoker is equipped with an adjustable feed coal feed control, built into the right hand crankshaft bracket of the stoker, to provide a great many rates of coal feed. This coal feed control may be manually operated if desired. Another lever is provided to control the feed of the coal from the master regulator. The hydraulically operated master regulator is also connected to individual vortex dampers of the blowers to vary the air in accordance with the fuel supply. The master regulator is also connected to the boiler uptake dampers in order to control the furnace draft.

Bituminous coal is received in cars on the railroad siding at the second floor level and discharged into a coal bunker having an approximate capacity of 250 tons. The coal is fed by gravity to the stoker hoppers through gates in the bunker. The ashes are taken from the boiler room in an elevator to the first floor or street level, to be loaded into trucks. The ashes can also be loaded in trucks taken direct to the boiler room on one of the truck elevators. A rubbish incinerator having a capacity of 2000 lb per hr is situated adjacent to the 200 hp boiler, and the heat from the incinerator is used under this boiler.

Steam at 100 lb pressure is required by the laundry, carpet and rug cleaning, lumber kilns, kitchens and other service departments, and, since this demand is quite variable, the boilers are operated at 125 lb pressure to have reserve pressure for sudden high steam demands.

Both boilers are provided with non-return valves and are connected to a main boiler header carrying steam at 125 lb. Steam at 100 lb pressure is supplied from the main header through a reducing valve and auxiliary header to the service departments requiring process steam at this pressure, and to all steam driven auxiliaries. A connection from the auxiliary header supplies steam reduced to 5 lb through a second reducing valve to the low pressure heating riser.

The condensate from the low pressure system is discharged into a receiving tank in the boiler room by a vacuum pump, and the high pressure returns are discharged directly into the receiver. A float valve connection to the receiver provides the necessary make-up water. Feed water is supplied to the boilers from this receiver by motor driven and steam driven boiler feed pumps.

Each boiler is equipped with a combination CO_2 recorder and flue gas temperature indicator, automatic damper regulator and feed water regulator.

The smoke ordinances of the city of Washington are very strict and require that all designs for boiler settings and incinerators be approved by the smoke inspector before installation. As a further precaution, smoke indicators actuated by photo-electric cells have been installed in the smoke breeching so that the fireroom attendant will have an audible alarm if excess smoke is produced.

The building was designed by Abbott, Merkt & Co., Inc., Industrial Engineers, of New York, and was constructed by Turner Construction Co. of Philadelphia. The Standard Engineering Co., Washington, installed the heating and ventilating systems.

Air Conditioning Service Engineer Tells How to Tune Up Installations

By Floyd M. Mayse*

ONE of the best means of eliminating trouble and keeping air conditioning maintenance expense to a minimum is the systematic inspection and preparation of the installation for proper operation each season. Sufficient time should be given the job to cover the various points outlined here.

Water Group

WATER PIPING—Check for leaks; piping disconnected for water drainage. *Remedy*—Replace leaky pipes; reconnect.

DRAIN PAN AND WATER SUMPS—Check for clogging, dirt. *Remedy*—Clean and flush out drain pans, water sumps, and drain connections. Lead and paint over with asphaltum paint all metal drain pans and water sumps. If a sump is made of wood, it should be cleaned thoroughly with water and after drying painted with two coats of hot linseed oil.

SPRAY NOZZLES—Check for dirt, proper settings. *Remedy*—Clean and adjust, whether on a cooling tower, evaporative condenser, or an air washer type system.

WATER PUMPS—Check for worn parts; leaky packing. *Remedy*—Should be disassembled, cleaned, and the worn parts replaced. Repacking is also recommended.

WATER VALVES—Check for dirt, worn parts. *Remedy*—Whether hand operated or automatic, water valves should be disassembled, cleaned, and worn parts replaced. Adjustment and operation should be checked before replacing in their final location.

Air Group

AIR DISTRIBUTION—Check for adjustment. *Remedy*—Adjust system. Set discharge grilles to obtain the best air distribution as determined by a smoke bomb test.

OUTSIDE AIR DUCT—Check for dirt. *Remedy*—Duct and its intake grille should be cleaned of dirt and trash. The outside air damper should be adjusted; it should be located where clean air will be drawn into the building at all times.

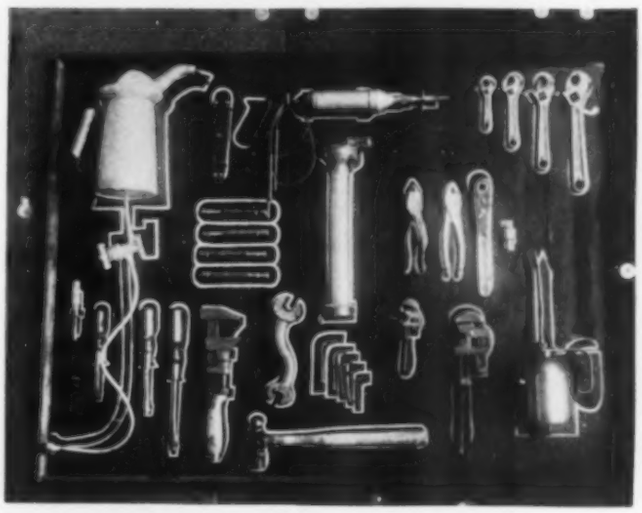
OTHER DUCT WORK—Check for air leaks, damper adjustment. *Remedy*—Repair air leaks and noises; adjust splitter dampers for correct air delivery into the branch ducts.

AIR FILTERS—Check for dirt or wear. *Remedy*—Whether in the outside air or return air to the units, filters should be cleaned or replaced as required.

AIR DAMPERS AND DAMPER MOTORS—Check for looseness, noise, operation. *Remedy*—Tighten and adjust.

FANS—Check for proper rotation, smoothness, lubrication. *Remedy*—Make necessary changes for proper rotation, looseness, and smoothness of operation. Fan bearings should be lubricated and aligned, if of the aligning type.

BELTS—Check for adjustment, excessive wear. *Remedy*—Compensate for wear and adjust. Set pulleys for proper alignment and tightness to minimize the wear on the belts and motor shaft.



Typical tool board for air conditioning work

Refrigerating Group

STRAINERS—Check for dirt, wear. *Remedy*—Whether water or refrigerant, strainers should be disassembled, cleaned, and replaced if necessary.

COILS—Check for dirt. *Remedy*—Clean heating and cooling coil surfaces by using water and either a stiff wire brush or a flexible metal bristle brush. Most of the dirt collected on the transfer surfaces will be found on the air inlet side of these coils.

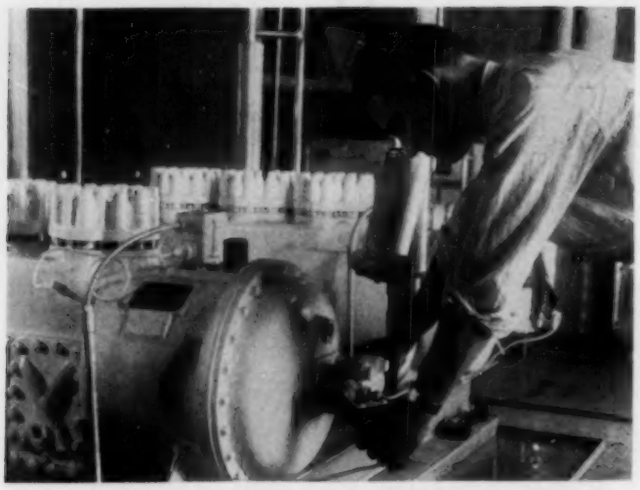
EXPANSION VALVES—Check for sluggish operation and improper adjustment. *Remedy*—Replace valve; adjust for proper superheat.

REFRIGERANT VALVES—Check for tightness, proper operation. *Remedy*—Adjust for positive closing and repack if necessary. If packless type valves are used, they should be replaced if found to be leaking through the diaphragm.

REFRIGERANT PIPING—Check for leaks. *Remedy*—Refrigerant piping should be leak tested with both halide

The suggestions given here for tuning up air conditioning installations are the results of the author's practical experience. A service engineer who spends most of his time in the field, he has a first-hand knowledge of the subject

*Air Conditioning Service Engineer, Westinghouse Electric & Mfg. Co.



Opening valves preparatory to starting equipment

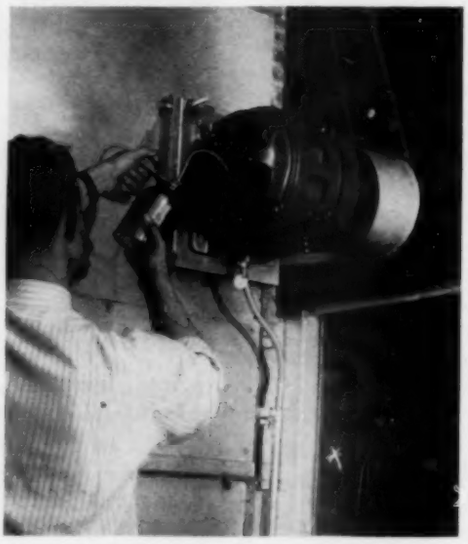
torch and soap bubbles under 150 lb per sq in. "Freon" pressure. Repair leaks.

REFRIGERANT QUANTITIES — Check for proper amounts. *Remedy*—After the system has been in operation for approximately one hour, refrigerant quantity may be determined by a sight flow indicator installed in the liquid line.

REFRIGERANT AND WATER PRESSURES—Check for proper values. *Remedy*—Install gages in the suction and liquid line of the refrigerant system and on the suction and discharge side of the water pump of the water system.

COMPRESSOR — After it has been started, check for oil quantity, oil pressure, and any irregular noises. *Remedy*—Make required repairs.

CONDENSER WATER TUBES—Check for scale deposit. *Remedy*—Clean any accumulated scale and water deposits. Some condensers may require cleaning by use of a recommended acid solution. For shell and tube type clean by standard wire tube cleaner.



Check lubrication of motors

CONDENSER WATER CONNECTION—Check for improper connection. *Remedy*—Connect for either parallel or series hook-up, as determined by the local conditions.

EVAPORATIVE CONDENSERS—Check for rust, dirt, scale. *Remedy*—Clean off rust, clean the finned condenser coils of all scale, dirt, and water deposits and paint the interior, including the fans and fan shaft, with an asphaltum paint with a finish coat of aluminum. Discharge duct should be cleaned of rust and repaired where necessary.

Electrical Group

MOTORS—Check for voltage and current, lubrication, direction of rotation, dirt, excessive noise, wear. *Remedy*—Open type motors should be oiled, and dirt and lint removed from inside the end bells. Bearings, windings, commutators, brushes, leads, end play and terminals should be replaced or repaired where necessary. Connections in the terminal box should be tight and well insulated. Worn commutators on dc drives should be sanded or turned.

WIRING—Check for incorrect wire size, loose connections, frayed insulation. *Remedy*—Tighten connections, replace worn wiring.

FUSES—Check for improper size. *Remedy*—Replace with correct size.

SWITCHES—Check for loose connections, worn parts. *Remedy*—Tighten connections.

LINE STARTER AND RELAY—Check for operation, contact pitting, proper overload setting, damaged arc boxes. *Remedy*—Smooth contacts if necessary, adjust relay, replace broken arc boxes.

CONTROL CIRCUITS—Check for broken wiring, loose connections, improper settings. *Remedy*—Tighten and adjust each control and safety device.

General Group

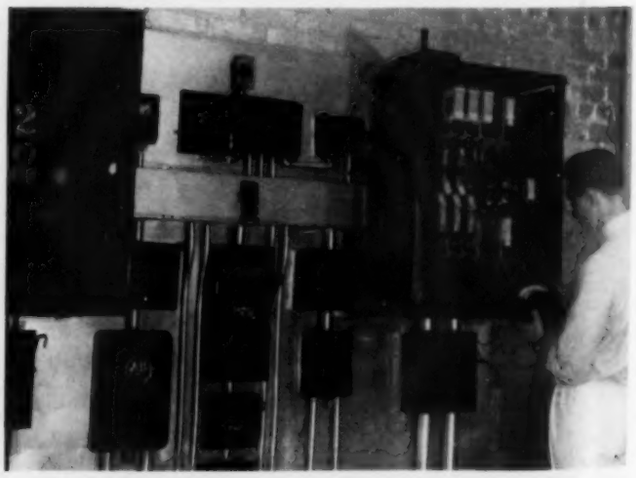
INSULATION—Check for leaks, deterioration. *Remedy*—Tighten and repair.

FOUNDATION—Check for sagging, excessive vibration. *Remedy*—Repair.

CLEANING AND PAINTING — Check for appearance, deterioration. *Remedy*—Wash and clean with a non-explosive solvent to remove grease and dirt. Equipment and all connecting lines should be painted in such a way as to make them stand out from all nearby machinery and building fixtures. In some cases it may be advisable to paint the floor and surrounding walls.

After the installation has been fully gone over, it is advisable to operate the system, rechecking the controls, interlocking devices, noises, water, air delivery, etc. After the system has operated normally for approximately five or six hours, it is advisable to run a capacity check on the complete system.

Checking electric circuits, controls and contacts



Committee on Grounding Hears Report on Research

By Howard S. Warren*

AT a recent meeting of the American Research Committee on Grounding, its technical subcommittee reported the progress made on the investigations of electrical grounding on pipes within buildings to the representatives of the 14 national organizations coöperating in this enterprise. The report covered the 18 investigations which have been made since the previous meeting about a year ago. Twenty-seven cases had been reported on previously. In addition to the field investigations a thorough review was made of the fundamental studies being carried on by the technical subcommittee.

The subcommittee has spent considerable time and effort investigating individual cases in the field where difficulty with either pipes or their contents might have been attributed to the effects of stray alternating current on the pipes as a result of electrical grounding. Several of these cases have added considerably to the committee's understanding of this general problem, while others seem to contribute relatively little in additional information.

One of the investigations involved a gas fire which resulted in serious damage to the building. In this case the gas meter and regulator valve had been disconnected for repairs, and the service plugged without shutting off the gas supply from the 5 lb street main. When the repairs had been completed and an effort made to reconnect the equipment, the gas ignited. The investigation showed that the grounding wire of an ac/dc radio set had been connected to the interior gas piping, which was very limited in extent and not grounded when disconnected from the service. It is probable that the normal current through the condenser of this radio set created a spark when the contact between the interior piping and the grounded street piping was made in the process of attempting to replace the meter. Another case involving a radio ground was investigated where a workman received a shock between disconnected house piping to which a radio set was connected and a grounded steam heating boiler.

Two of the most interesting cases investigated by the committee involved adjacent residences where copper tubing had failed within 2½ years of installation. In the first residence two sections of interior tubing had failed and some other sections were badly pitted on the inside. Still other sections carrying the same water at approximately the same temperature were in excellent condition. In the second house only one section of the tubing had failed. In the first house no stray current was found on any of the interior piping, but in the second house there was stray current on the interior piping. The copper tube service pipes to both houses, which carried unbalanced neutral currents, were found to be in good condition. Analyses of the various sections of pipe removed from these houses revealed that they conformed to the ASTM specifications for copper tubing—99.9 per cent pure copper, free from cuprous oxide. At the present time the committee is studying the effects of the variables within the 1/10 of 1 per cent impurities permitted by the standard specifications. The analyses so far indicate that the small differences in some of these impurities may go far to explain the difference in corrosion of these pipes. In this case the water involved could be classified as a border line water as far as corrosiveness is concerned.

A working group of the technical subcommittee making an investigation

In several cases blue staining of plumbing fixtures was involved. In two of these, the houses were new and the staining disappeared after several months. In other cases corrosive water and various combinations of different metals in the piping system seem to be contributing factors.

Rather extensive studies have been made of the effects on electrical grounding of the use of cement-asbestos mains instead of metallic mains. In one case the water company was planning to replace an 800 ft dead-end cast iron main with a cement-asbestos main. This gave an opportunity for a study of conditions with the two types of mains. In another location the committee made an investigation in an area where cement-asbestos mains were extensively used. With cement-asbestos mains it

The objectives of the studies being made by the American Research Committee on Grounding were described by Mr. Warren in the August, 1938, HEATING, PIPING AND AIR CONDITIONING. . . . The present report tells of the work done since the previous meeting of the committee about a year ago, and will bring the reader up to date on the group's investigations

*Consulting Electrical Engineer and Chairman, American Research Committee on Grounding.

Large split core transformer as used for measuring current in water main

was found that the individual water service might have a relatively high resistance to ground and that the main itself was essentially an insulator. In practically every case through contacts within the house, other structures such as gas piping, sewer piping, etc., contributed to lowering the overall resistance to ground. However, in several situations a low resistance to ground was obtained only through the paralleling of the individual grounds of many customers through the power system neutral.

Fundamental laboratory studies of the corrosion of metals with superimposed alternating and direct current on galvanic cells are being continued. Apparatus has been developed such that instantaneous polarization effects can be studied. To date the tests have been made with copper electrodes in 1/10 normal sodium chloride solution. The studies have included various values of alternating and direct current, both separately and in combination. The tentative results obtained are:

1) For d-c tests the oxygen content of the electrolyte has a very important influence on the polarization phenomena.

2) The presence of alternating current has a negligible effect on the electrode potential at the current densities used in the test.

3) Alternating current superimposed on direct current has negligible effect on the polarization caused by the direct current whether the alternating current is applied before or after the direct current.

Before final conclusions can be drawn, considerable more laboratory work will be required. Further laboratory work will include electrodes of metals other than copper, other electrolytes, electrodes of different sizes, and electrodes of dissimilar metals.

The committee is also planning to conduct further tests on pipes carrying alternating and dirrect current. It is planned to control carefully the conditions of these tests so as to eliminate, as far as possible, experimental error. It is expected to determine any effects on the water samples obtained from the pipes by taste and odor rating, and by spectrographic or chemical methods of analysis. Any effects on the pipes will also be observed.

In the investigations to date the committee has found various unsafe and unsatisfactory conditions in electric wiring, water pipes, gas pipes, etc. Many of these were in violation of established good practice. However, to date the committee has been unable to find any case where electrical grounding was shown to be a material factor in causing damage to water pipes or their contents.

The problem of obtaining and maintaining a satisfactory grounding attachment to the water pipe is noted in many instances. The most promising suggestion towards solving this problem is the installation at the time of construction of a special fitting in the water pipe near the water service entrance, to which a lug could be bolted by the electrician. The use of a jumper between pipes when gas or water meters are disconnected would substantially reduce the possibility of sparking or shock to workmen.

Galvanic cells set up by dissimilar metals in the piping system appear to be an important factor in this problem.

Minor impurities within the present specifications for copper used in piping or tubing appear to be a possible factor in the problem that is variously referred to as "blue stain" or "green water."

Where cement-asbestos mains are used, the importance of the individual grounding connections becomes much greater. Also, the importance of a continuous multi-grounded conductor increases in order to insure an overall low resistance ground for each service.

Library "Treasure Room" Air Conditioned

By **Harold S. Birkett***

AIR conditioning will be used to preserve rare manuscripts and valuable books at the new Central Public Library, which will serve 2½ million residents of Brooklyn, New York.

In the preservation of important books, prints and folios, it is desirable to maintain the room air at a constant humidity—for it is variation in the water vapor or moisture content of the surrounding air which is the principal cause of deterioration of printed matter. Low humidities cause bindings and pages to dry prematurely, whereas high humidities promote the formation of fungus growth. Further, the paper, leather, glue, sizing and other materials of which books are made expand and contract in relation to the wetness or dryness of the air, and when this occurs a straight mechanical breakdown of volumes is fostered.

In Brooklyn's new library, nearing completion, all works to be guarded will be kept in a "Treasure Room" in which the relative humidity of the air will be kept constant at 40 per cent by means of gas fired silica gel air conditioning equipment, the principal function of which is ventilation and humidity control. Temperature control or cooling, although to be provided, is incidental. In addition, the air conditioning system will embrace the librarian's office and study and certain trustees' and committee rooms.

*The Brooklyn Union Gas Co.

Fuel Consumption Estimating Formulas Given in Housing Authority Bulletin

THE appendix to the United States Housing Authority's bulletin on the heating of low rent housing projects, comprising fuel consumption formulas, is presented in the following paragraphs:

1) Explanatory Note: EDR (Equivalent Direct Radiation) as used herein is the total heating surface used for space heating, including ferrous or non-ferrous radiators or convectors, pipe coils, exposed risers, unit heaters, etc., 1 sq ft EDR being equal to a heat emission of 240 Btu per hr.

2) STEAM HEATING

a) Base formula for coal used for space heating:

$$\frac{\text{EDR} \times \text{Fuel Factor} \times \text{Annual Degree Days}}{2000 \times \text{Efficiency}} = \text{Tons per yr}$$

For efficiencies in central or group plants, the following are suggested: 60 per cent—hand firing; 65 per cent—stoker fired (low pressure); 70 per cent—stoker fired (high pressure).

Fuel factor = Lb of coal per sq ft radiation per degree day for corresponding design temperature and 100 per cent efficiency (see Table 1).

b) Base formula for gas used for space heating:

$$\frac{\text{EDR} \times \text{Fuel Factor} \times \text{Annual Degree Days}}{1000 \times \text{Efficiency (use 75 per cent)}} = \text{M cu ft gas per yr}$$

Fuel factor = Cu ft gas per sq ft radiation per degree day for corresponding design temperature and 100 per cent efficiency (see Table 2).

c) Base formula for oil used for space heating:

$$\frac{\text{EDR} \times \text{Fuel Factor} \times \text{Annual Degree Days}}{\text{Efficiency (use 75 per cent)}} = \text{Gal oil per yr}$$

Fuel factor = Gal oil per sq ft radiation per degree day for corresponding design temperature and 100 per cent efficiency (see Table 3).

d) For formula for estimating district steam consumption, see paragraphs 9a, b and c.

e) For all space heating estimates during continuous operation, 20 per cent should be added to the fuel consumption if zone control is omitted where the design temperature is 0 F or higher, and 15 per cent where the design temperature is less than 0 F. Twelve per cent should then be added to the fuel consumption to compensate for line losses in high pressure central plants and 10 per cent in low pressure central or group plants.

f) The same base formulas outlined hereinbefore can be applied to individual building and individual dwelling unit plants for space heating estimates. The individual building plants scheme can also include a boiler plant serving two or three small buildings. The following efficiencies are suggested: hand firing, bituminous coal—44 per cent; hand firing, anthracite coal—52 per cent; stoker fired, bituminous coal—50 per cent; stoker fired, anthracite coal—55 per cent; gas—70 per cent; oil—65 per cent.

A USHA aided project in Buffalo, N. Y.

g) Five per cent should then be added to the fuel consumption to compensate for line losses in individual building plants.

h) The line loss percentages suggested herein should be used for preliminary calculations only.

3) HOT WATER HEATING

a) The EDR should be transposed in terms of its equivalent in steam radiation. The base formulas including compensation for line losses and for omission of zone control or other regulation set forth in paragraph 2 can then be applied.

b) Formula for transposing Hot Water EDR to Steam EDR:

$$\frac{\text{EDR (Hot Water)} \times \text{Btu per sq ft (Hot Water)}}{240} = \text{EDR (Steam)}$$

4) WARM AIR HEATING (Forced and Gravity)

a) The Btu loss per hr should be transposed in terms of its equivalent in steam equivalent direct radiation by dividing by 240 (see explanatory note, paragraph 1). The base formulas as outlined in paragraph 2 can then

The appendix to the United States Housing Authority's bulletin on the heating of low rent housing projects comprises fuel consumption formulas, and is published here. . . . The bulletin's introductory paragraphs, and Part 1 on economic analysis, appeared in the June HPAC; Part 2, giving recommendations and suggestions for design of heating plants, was published in July; and Part 3, on preparing the contract documents, was presented in the August issue

be applied for the steam or hot water coiled forced warm air unit heater system, for the individually fired forced or gravity heater systems. In all cases compensation should be made for line losses, and for omission of zone control or other regulation.

5)

a) Examples of use of space heating formulas:

Assume 15,000 steam EDR (low pressure—central plant—zone controlled). 5000 degree days (annual); —10 to 70 F design range; 12,000 Btu per lb of coal; 1000 Btu per cu ft of gas; 140,000 Btu per gal of oil.

b) Coal

$$\frac{15{,}000 \text{ EDR} \times 5000 \text{ DD} \times 0.007}{1300} = 404 \text{ tons coal per yr}$$

Plus 10 per cent line losses = 404 + 40.4 = 444.4 tons per yr.

c) Gas

$$\frac{15{,}000 \text{ EDR} \times 5000 \text{ DD} \times 0.084}{750} = 8400 \text{ M cu ft gas per yr}$$

Plus 10 per cent for line losses = 8400 + 840 = 9240 M cu ft per yr.

d) Oil

$$\frac{15{,}000 \text{ EDR} \times 5000 \text{ DD} \times 0.0006}{0.75} = 60{,}000 \text{ gal oil per yr}$$

Plus 10 per cent for line losses = 60,000 + 6000 = 66,000 gal oil per yr.

6)

a) Base formula for fuel used for domestic hot water heated through steam coiled generators. (No. of dwelling units × 60 gal per day × 1.1 lb steam per gal × 365 days per year × 960 Btu per lb steam) all divided by Btu per unit of fuel used × 0.50 (50 per cent assumed efficiency) = units of fuel used per year.

b) Coal

$$\frac{\text{DU} \times 60 \times 1.1 \times 365 \times 960}{\text{Btu per lb coal} \times 0.50 \times 2000 \text{ lb per ton}} = \text{tons coal per yr}$$

or in simplified form

$$\frac{\text{DU} \times 23{,}127}{\text{Btu per lb coal}} = \text{tons coal per yr}$$

c) Gas

$$\frac{\text{DU} \times 60 \times 1.1 \times 365 \times 960}{\text{Btu per cu ft gas} \times 0.50 \times 1000} = \text{M cu ft gas per yr}$$

or in simplified form

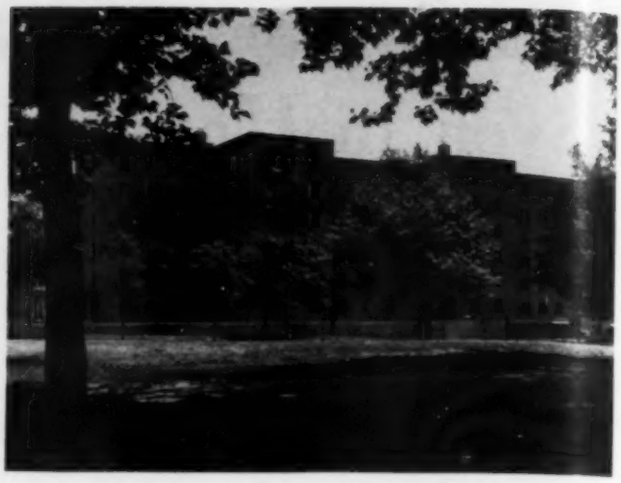
$$\frac{\text{DU} \times 46{,}254}{\text{Btu per cu ft gas}} = \text{M cu ft gas per yr}$$

d) Oil

$$\frac{\text{DU} \times 60 \times 1.1 \times 365 \times 960}{\text{Btu per gal oil} \times 0.50} = \text{gal oil per yr}$$

Example for Degree Days Tabulation

1	2	3	4	5	6	7	8
Month	Degree Days	Per Cent of Total	Heating—M Lb Steam		Dom. H. W.—M Lb Steam		Total M Lb Steam Cols. 5 Plus 7
				Col. 4 Plus 12% Line Losses		Col. 6 Plus 12% Line Losses	
Jan.	682	23.5	5180	5802	1212	1357	7159
Feb.	557	19.0	4230	4738	1212	1357	6095
Mar.	388	13.5	2950	3304	1212	1357	4661
Apr.	132	4.5	1000	1120	1212	1357	2477
May					1212	1357	1357
June					1212	1357	1357
July					1212	1357	1357
Aug.					1212	1357	1357
Sept.					1212	1357	1357
Oct.	96	3.5	730	818	1212	1357	2175
Nov.	396	13.5	3000	3360	1212	1357	4717
Dec.	639	22.5	4860	5443	1212	1357	6800
Total	2890	100.0	21950	24585	14544	16284	40869



Red Hook development in New York City

or in simplified form

$$\frac{\text{DU} \times 46{,}254{,}000}{\text{Btu per gal oil}} = \text{gal oil per yr}$$

e) For domestic hot water heating fuel consumption where underground heat distribution for this purpose is required, the same compensation for line losses may be applied as noted in paragraph 2e. No heat line losses need be figured where hot water generators are installed in boiler plant buildings.

f) The daily gallon consumption and line loss percentages suggested herein should be used for preliminary calculations only.

g) For formulas for estimating district steam consumption, see paragraphs 9d, e and f.

7)

a) Examples of use of domestic water heating formulas:

Assume 100 dwelling units (central plant); 12,000 Btu per lb coal; 1000 Btu per cu ft gas; 140,000 Btu per gal oil; hot water generators are in boiler plant (no heat line losses figured).

b) Coal

$$\frac{100 \times 23{,}127}{12{,}000} = 192.8 \text{ tons per yr}$$

c) Gas

$$\frac{100 \times 46{,}254}{1000} = 4625.4 \text{ M cu ft gas per yr}$$

d) Oil

$$\frac{100 \times 46{,}254{,}000}{140{,}000} = 33{,}038 \text{ gal oil per yr}$$

8)

a) Base formula for fuel used for domestic hot water heating through individual heaters (No. of dwelling units × 40 gal per day × 8⅓ lb of water per gal × 100 F temperature rise × 365 days per year, all divided by Btu per unit of fuel used × efficiency = units of fuel used per year). The water temperature rise is basic only. Initial water temperatures may vary dependent upon locality.

b) Coal

$$\frac{\text{DU} \times 40 \times 8\tfrac{1}{3} \times 100 \times 365}{\text{Btu per lb coal} \times \text{eff.} \times 2000} = \text{tons coal per yr}$$

Table 1—Fuel Factor Table—Lb Coal per Sq Ft per Degree Day at 100 Per Cent Efficiency

Btu per Lb	— 20 F	— 10 F	— 5 F	0 F	+ 5 F	+ 10 F	+ 20 F
11,000	0.00678	0.00764	0.00813	0.00872	0.00938	0.01018	0.01222
11,500	0.00648	0.0073	0.0078	0.00835	0.00899	0.00975	0.017
12,000	0.00622	0.007	0.00746	0.008	0.00862	0.00933	0.0112
12,500	0.00596	0.00672	0.00716	0.00767	0.00826	0.00896	0.01075
13,000	0.00574	0.00646	0.00689	0.00738	0.00795	0.00862	0.01034
13,500	0.00562	0.00622	0.00664	0.00712	0.00767	0.0083	0.00996
14,000	0.00532	0.006	0.00639	0.00685	0.00737	0.0080	0.0096
14,500	0.00514	0.00578	0.00617	0.00662	0.00712	0.00772	0.00926

Table 2—Fuel Factor Table—Cu Ft Gas per Sq Ft per Degree Day at 100 Per Cent Efficiency

Btu per Cu Ft	— 20 F	— 10 F	— 5 F	0 F	+ 5 F	+ 10 F	+ 20 F
500	0.14933	0.168	0.179	0.192	0.2065	0.224	0.2688
1000	0.07466	0.084	0.0896	0.096	0.1033	0.112	0.1344
1040	0.07180	0.08077	0.0862	0.0923	0.0994	0.10769	0.12923

Table 3—Fuel Factor Table—Gal Oil per Sq Ft per Degree Day at 100 Per Cent Efficiency

Btu per Gal	— 20 F	— 10 F	— 5 F	0 F	+ 5 F	+ 10 F	+ 20 F
140,000	0.000533	0.0006	0.00064	0.0006857	0.00737	0.0008	0.00096
148,000	0.000505	0.000568	0.000606	0.000649	0.0007	0.000757	0.0009081

Table 4—Fuel Factor Table—Lb Steam per Sq Ft per Degree Day at 100 Per Cent Efficiency

— 20 F	— 10 F	— 5 F	0 F	+ 5 F	+ 10 F	+ 20 F
0.0778	0.0875	0.09334	0.1	0.1077	0.11667	0.14

Use 35 per cent efficiency, or in simplified form

$$\frac{\text{DU} \times 17{,}381}{\text{Btu per lb coal}} = \text{tons coal per yr}$$

c) Gas

$$\frac{\text{DU} \times 40 \times 8\tfrac{1}{3} \times 100 \times 365}{\text{Btu per cu ft gas} \times \text{eff.} \times 1000} = \text{M cu ft gas per yr}$$

Use 55 per cent efficiency for flue type heater, or in simplified form

$$\frac{\text{DU} \times 22{,}120}{\text{Btu per cu ft gas}} = \text{M cu ft gas per yr}$$

d) Oil

$$\frac{\text{DU} \times 40 \times 8\tfrac{1}{3} \times 100 \times 365}{\text{Btu per gal oil} \times \text{eff.}} = \text{gal oil per yr}$$

Use 50 per cent efficiency, or in simplified form

$$\frac{\text{DU} \times 24{,}332{,}300}{\text{Btu per gal oil}} = \text{gal oil per yr}$$

e) Electricity

$$\frac{\text{DU} \times 40 \times 8\tfrac{1}{3} \times 100 \times 365}{\text{Btu per kwhr} \times \text{eff.}} = \text{kwhr per yr}$$

Use 77 per cent efficiency, Btu per kwhr = 3412, or in simplified form

$$\text{DU} \times 4631 = \text{kwhr per yr}$$

f) The daily consumption suggested herein should be used for preliminary calculations only.

9)

a) Base formula for district steam used for space heating:

$$\frac{\text{EDR} \times 96 \text{ Btu} \times \text{monthly degree days} \times 70 \text{ F range}}{960 \text{ Btu} \times 1000 \times \text{design temp. range}} = \text{M lb steam}$$

for the particular month that the degree days are selected.

If no zone control is provided, 20 per cent should be added where the design temperature is 0 F or higher and 15 per cent where the design temperature is less than 0 F. Twelve per cent should be added for line losses, if these losses are to be absorbed by the project.

Note: The estimate for each month must be computed separately, as district steam companies render bills monthly.

b) Simplified formula for district steam (space heating):

$$\frac{\text{EDR} \times \text{monthly degree days} \times \text{factor (from Table 4)}}{1000} = \text{M lb steam per mo}$$

Plus compensation for line losses and omission of zone control.

c) The yearly total can be figured from this formula by substituting the annual degree days in lieu of the monthly degree days. Compensation for line losses and for omission of zone control should be made.

d) Base formula for district steam used for domestic water heating:

$$\frac{\text{DU} \times 60 \times 1.1 \times 365}{12 \times 1000} = \text{M lb steam per mo}$$

The same line losses as applied to heating should be added where underground heat distribution for this purpose is required.

e) Simplified formula for district steam for domestic hot water:

$$\text{DU} \times 2.007 = \text{M lb steam per mo}$$

plus compensation for line losses.

f) Simplified formula for district steam for domestic hot water (annual consumption):

$$\text{DU} \times 24.084 = \text{M lb steam}$$

plus compensation for line losses.

g) Example: 604 DU; 76,000 EDR; 0 F — 70 F design temperature range; zone control; hot water generators located in basement of certain buildings.

$$\text{Space heating} = \frac{76{,}000 \times \text{degree days} \times 0.1}{1000} = \text{M lb steam}$$

(where 0.1 = fuel factor)

M lb steam = 7.6 × degree days (see degree day tabulation), plus 12 per cent for line losses.

Domestic hot water = 604 DU × 2.007 = 1212.2, plus 12 per cent for line losses.

10) Note: Origin of Constant 96 Used in Tables 1, 2 and 3.

About 1926 a large number of actual fuel consumption figures (in cu ft of gas per degree day per sq ft of steam radiation) was collected by gas utilities. These figures, averaged, were reduced to a figure (at 0 to 70 F design conditions) of 0.218 cu ft of 535 Btu per cu ft gas per sq ft of steam radiation per degree day, or, in other words, an input of 116.6 Btu per sq ft of steam radiation per degree day. Here was a figure which was actually obtained in the field and from which could be calculated a theoretical figure at 100 per cent efficiency. It was assumed that these gas heating plants were operating at an overall efficiency of 82.5 per cent, so that if the plants had been operating at an efficiency of 100 per cent they would have required only 116.6 × 0.825 or 96 Btu per sq ft of steam radiation per degree day.

Consequently the constant 96 Btu per sq ft of steam radiation per degree day is the basis of all fuel constants used in the accompanying tables.

[*This concludes publication of this bulletin.*]

A 36 p. report, "The Treatment of Coal with Oil and Other Petroleum Products," by J. M. Pilcher and Ralph A. Sherman, fuel research engineers of Battelle Memorial Institute, Columbus, Ohio, has been issued as Technical Report No. VI of Bituminous Coal Research, Inc. It is based on the results of a 2½ year program of research at Battelle.

How Trouble With a Vacuum Pump Was Corrected With Lift Fittings

By W. F. Fischer*

"I DON'T understand why the old vacuum pump should race and clatter the way it does!" the boiler attendant said. "It always ran slow and quietly before we moved it to the boiler room."

The pump was a steam driven double acting vacuum return line heating pump that had originally been situated in the drying room shown at the left of Fig. 1. The water cylinder of the pump had an 8 in. bore, a 9 in. piston stroke, and a displacement of 0.26 cu ft per stroke. The pump received the condensate from 15,000 sq ft of cast iron wall radiation and pipe coils by gravity, and delivered it through the sloping return line in the pipe tunnel to the boiler room.

The pump had been moved to the boiler room, where it could be given more attention, and it then began giving trouble. Some one suggested that the radiation traps must be leaking steam, and recommended that a "spray chamber" be placed in the suction line on the inlet side of the pump to condense the steam and vapor before it reached the pump. The spray device helped to some extent, but used considerably more water than was required to make up the small amount lost from the heating system. When the amount of water was cut down, the pump started to race and pound again as badly as before.

"Your main trouble," I told the attendant, after looking over the heating system, "is that you are trying to lift hot water, at a temperature very close to 212 F, by suction and pull it up to the boiler room through a long sloping line."

"But the pump does lift the water!" he said. "It doesn't race and clatter steadily—only at certain times."

"The pump doesn't actually lift the water at all," I remarked, "it simply creates a partial vacuum in the suction line, and the pressure at the other end forces the water to flow to the pump. The hot condensate starts releasing steam or vapor, as well as air, and other gases from impurities in the water, and these elastic fluids expand rapidly under a high degree of vacuum, and have to be removed before the water can be lifted by the pressure at the other end of the suction line. The vapor, air and other gases arising from the water tend to break the vacuum, and thus cause the pump to race and clatter—before it can be controlled by the governor. The hot water can be lifted to the pump inlet only after it has cooled to a point where it no longer gives off vapor in sufficient quantities to break the vacuum in the suction line. Hot water should, of course, flow to the pump by gravity."

"Does that mean that we'll have to put the pump back in the basement again?" he asked.

"No, you can leave it where it is," I said. "We'll simply change the suction line, put in some 'lift fittings', and make the condensate flow to the pump by gravity."

*The author has been connected with several of the large engineering firms during the past 25 years.

Why the Pump Gave Trouble

By referring to Fig. 1, we see that the pump, as installed in the boiler room, had to lift the returns to a vertical height of 8 ft 2 in. over a horizontal distance of 161 ft 6 in. (Fig. 2) to the inlet of the pump.

The vertical height in feet to which pure water standing in an open vessel exposed to the atmosphere can be lifted by suction depends on the pressure of the atmosphere, the weight of a column of the water one foot high and one inch square at its base, and the absolute vapor pressure of the water in pounds per square inch.

Let L represent the suction lift, ft;
W the weight of water at the given temperature, lb per cu ft;
P the absolute vapor pressure, lb per sq in.;
A the absolute pressure of the atmosphere, lb per sq in.

Then

$$L = \frac{144\,(A-P)}{W} \qquad [1]$$

Solving this equation for P, we get

$$P = A - \frac{WL}{144} \qquad [2]$$

Since P and W both vary with the temperature of the water, we must approximate the value of W in order to find the value of P from equation 2. An average value for W for water at temperatures ranging from 170 to 212 F is 60.25 lb per cu ft, which is accurate enough for the purpose.

Taking $A = 14.7$ lb per sq in. at sea level, we can write as an approximate formula

$$L = \frac{14.7-P}{0.418} \qquad [1a]$$

$$P = 14.7-0.418L \qquad [2a]$$

From Fig. 1, $L = 8$ ft 2 in., or 8.167 ft; hence $P = 14.7 - (0.418 \times 8.167) = 11.29$ lb per sq in. absolute.

The temperature corresponding to this vapor pressure is found in the steam tables (Marks and Davis, used by the writer) to be 199 F. We see, therefore, that the condensate in the suction line would have to cool to this temperature before it can be lifted to the pump inlet by suction.

If the initial temperature of the condensate is 212 F. and no heat is lost in the release of pressure from 14.7 to 11.29 lb per sq in. absolute—there will be liberated from each pound of the condensate under vacuum $212 - 199 = 13$ Btu, which furnishes the latent heat

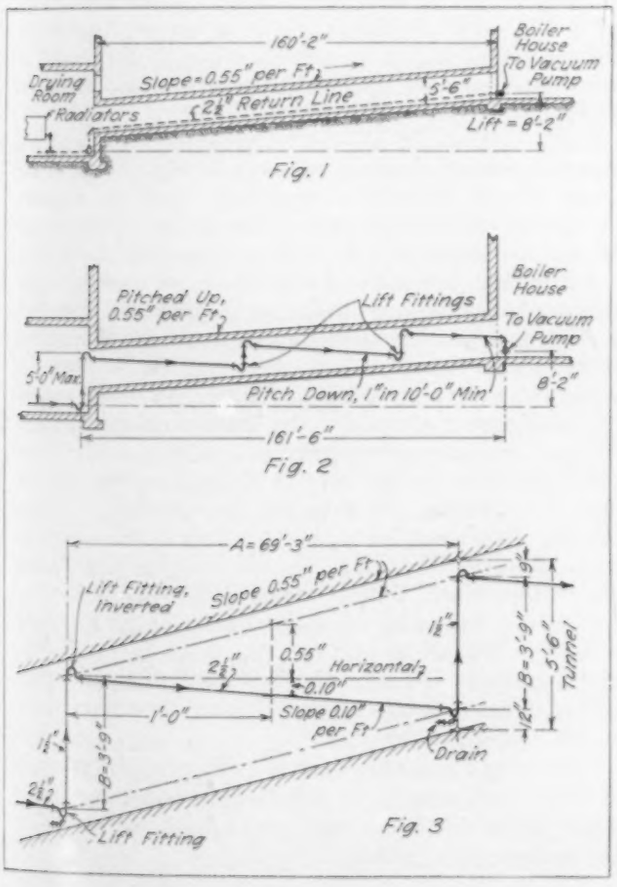
necessary to flash a portion of the water to steam or vapor at the lower temperature of 199 F.

The latent heat of evaporation for saturated steam at 11.29 lb per sq in. absolute and 199 F, is (from the steam tables) 978.4 Btu per lb, so that $13 \div 978.4 = 0.0133$ lb of vapor will be released from every pound of the condensate going to the pump, leaving 0.9867 lb as water. In other words, 1.33 per cent of the condensate must be lifted as vapor and 98.67 per cent as water. This might appear to be too small a quantity of vapor to worry about, but when we consider that the specific volume of steam at the given pressure and temperature is 34.3 cu ft per lb, the pump would have to lift and remove $0.0133 \times 34.3 = 0.4562$ cu ft of vapor from every pound of water going to the pump.

Water at 199 F weighs 60.1 lb per cu ft, so that 0.9867 lb of water at this temperature will occupy a space of $0.9867 \div 60.1 = 0.01642$ cu ft. By volume, therefore, the ratio of vapor to water will be 0.4562/0.01642 or 27.8 to 1. This shows that the pump would have to handle 27.8 cu ft of vapor for every cubic foot of water going to the pump.

The volumetric displacement rate of the water cylinders of vacuum return line pumps is usually made from eight to ten times greater than would actually be required to handle the volume of water alone. In the present case, however, the displacement rate would have to be over 27 times as great.

Fig. 1—Longitudinal section through pipe tunnel showing return line as originally installed . . . Fig. 2—Return line with lift fittings . . . Fig. 3—How to determine maximum center to center dimensions of fittings in sloping tunnel or trench



When a vacuum return line heating pump was moved to a new location, trouble was caused by its racing and clattering, due to the pump's trying to handle hot water at a temperature close to 212 F and pull it up through a long sloping suction line. . . . The author explains why the trouble occurred, and shows how it was corrected by installing lift fittings. He also gives information on determining the dimensions and spacings of lift fittings, and sketches

Since the pump serves 15,000 sq ft of cast iron wall radiation and pipe coils, and each sq ft of surface gives off 300 Btu per hr, there will be a total of $15{,}000 \times 300 = 4{,}500{,}000$ Btu given off per hr, or 75,000 Btu per min. This means that $75{,}000 \div 970.4 = 77.3$ lb per min of steam at 212 F will be required (970.4 = latent heat of steam at 212 F). Consequently, this amount of condensate must be handled by the vacuum pump every minute. The pump, therefore, will have to handle $77.3 \div 60.1 = 1.29$ cu ft of water, and $77.3 \times 0.4562 = 35.26$ cu ft per min of vapor, or a total of 36.55 cu ft per min.

As before mentioned, the water cylinder of the pump has a volumetric piston displacement of 0.26 cu ft per stroke. Taking the pump slip as $\frac{1}{6}$ the displacement, this reduces to $\frac{5}{6} \times 0.26 = 0.217$ cu ft per stroke.

The pump, therefore, will have to make $36.55 \div 0.217 = 168$ strokes per minute in order to handle the water and vapor that comes to it. Then, since the length of stroke is 9 in. or 0.75 ft, the piston speed will have to be $168 \times 0.75 = 126$ fpm.

For small pumps such as this it has been recommended that piston speed in feet per minute should not exceed 20 times the square root of the stroke in inches; that is, $20\sqrt{9}$ or 60 fpm, in the present case. This means that the piston would have to travel more than twice as fast as the maximum speed recommended for such pumps.

Quantity of Water to Condense Vapor

Since the pump would have to handle $0.0133 \times 77.3 = 1.028$ lb per min of vapor at a temperature of 199 F, and the latent heat of evaporation at this temperature is (from the steam tables) 978.4 Btu per lb, there will be $1.028 \times 978.4 = 1006$ Btu which must be absorbed by the spray water every minute, in order to condense the vapor before it reaches the pump.

The quantity of spray water required depends, of course, on the initial and final temperatures of the water. If the initial temperature is 60 F, and the final temperature, before it mixes with the condensate, 120 F, each pound of water will absorb 60 Btu and there will be required $1006 \div 60 = 16.8$ lb per min of water, or 1008 lb per hr. Even if the final temperature of the water were to reach the same temperature as the vapor itself (199 F), there would still be required $1006 \div (199 - 60) = 7.24$ lb per min or 434 lb per hr, which is quite expensive when purchased at city meter rates over the complete heating season, and then allowed to go to waste carrying a considerable quantity of heat with it.

The above calculations, of course, give approximate results only, but tend to show why trouble is experienced

in cases of this kind. Some vapor would, of course, be condensed by the radiation and convection of heat from the suction line, but this is in turn offset by the amount of steam or vapor that passes the radiator traps every time they open to discharge condensate from the radiators or pipe coils. The pump also has to handle a certain amount of air that leaks in under vacuum, as well as air and other gases given off from the water when heated. It all comes back to the vacuum pump.

Lift Fittings Correct Trouble

In the light of the foregoing explanation, the trouble with the vacuum pump was corrected by the use of lift fittings. Fig. 2 shows how the suction line in the pipe tunnel was changed by installing the lift fittings and grading the return line downward in the direction of flow, so that the condensate would flow to the pump by gravity. The vertical lift pipe is made one or two sizes smaller than the horizontal return, and has its lower end submerged in the well of the lower lift fitting, below the level of the lower horizontal pipe. A similar fitting, used in an inverted position, is placed at the upper end of the lift pipe, as shown in the illustration, to prevent the water from draining back again after it has reached the top. The condensate accumulating in the lower well seals the end of the vertical pipe and thus reduces the vacuum slightly on the inlet side of the fittings. When this occurs the pressure builds up in the return line and forces the water up through the lift pipe in piston-like slugs. The ascending water, carrying air, vapor and other gases with it, is prevented from falling back when it reaches the top, and then flows by gravity through the upper horizontal pipe and into the well of the next lift fitting, where the process is again repeated.

Lifts of 6 ft or over should not be made in one step, but should be divided into two or more shorter steps (not necessarily of equal height), depending on the total height to which the condensate is to be raised. Several short lifts are, as a rule, easier on the vacuum pump than one high lift, as the water is seldom being lifted in all vertical steps at the same time; one step may be lifting while another is discharging, and while the condensate is accumulating in the well of a third step. The tendency is, however, to keep a steady stream of condensate, vapor, and air moving in the direction of the vacuum pump.

If the return line is laid on the floor, and the pump set at a higher level, a pair of lift fittings should be used at a point close to the pump to lift the condensate above the inlet connection and permit it to flow into the pump by gravity. The pump will then be getting a fairly steady mixture of water, vapor and air, and will not have to exhaust a large volume of vapor before the water starts flowing to the pump. All horizontal pipes should be graded in the direction of flow, at least ½ in. in 10 ft.

Where the lifts are to be made in a pipe tunnel (or trench) as shown in Fig. 2, the height of each lift is governed by the available head room in the tunnel, but should not exceed from 5 to 6 ft as a maximum. In the present case the tunnel has an upward slope of 0.55 in. per ft in the direction of flow and this must be added to the downward slope of the return line, 0.10 in. per ft, in order to determine the maximum center to center

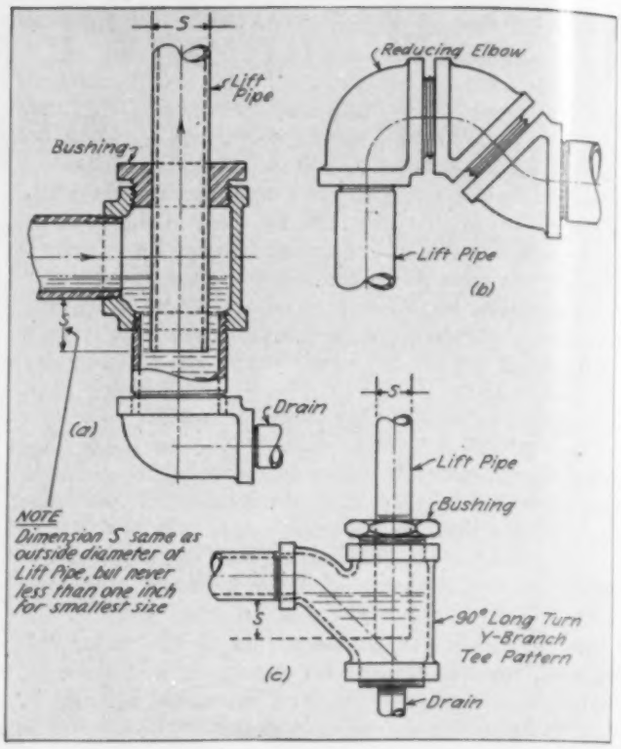


Fig. 4—Use of screwed pipe fittings in making lift fittings

of lifts. As shown in Fig. 3, the clear head room in the tunnel is 5 ft 6 in. and the vertical center to center dimension B of lift fittings is 3 ft 9 in., or 45 in. (this can be increased slightly depending on the clearance required). The maximum center to center dimension A of vertical lifts is therefore $A = 45 \div 0.65 = 69.25$ or 69 ft 3 in. center to center. This, of course, determines the number of steps that must be made in the tunnel to keep the return line graded downward in the direction of flow.

Each vertical lift should be provided with a drain connection at the base as indicated in Fig. 3, and these drains should be opened occasionally, with the vacuum pump shut down, in order to blow out the foreign matter that accumulates in the fittings from time to time.

Lift fittings can be purchased which are correctly proportioned for the purpose for which they are intended.

Fig. 4 shows lift fittings constructed from ordinary screwed pipe fittings. In *a* the lower or lift fitting is made from a reducing tee, a bushing, a short nipple, and a reducing elbow for the drain connection. The lift pipe, as before mentioned, should be smaller than the return line by one or two sizes, depending on the size of the return. A reducing 90 deg elbow and two 45 deg elbows can be used at the upper end of the lift pipe, to prevent the water from draining back again into the vertical pipe. This is shown in *b*. The writer has made these fittings by using a reducing 90 deg long turn Y-branch, bushed on the run, as shown in *c*. A long running thread is cut on the lower end of the lift pipe, which is screwed into the bushing to the proper depth to seal the end of the pipe.

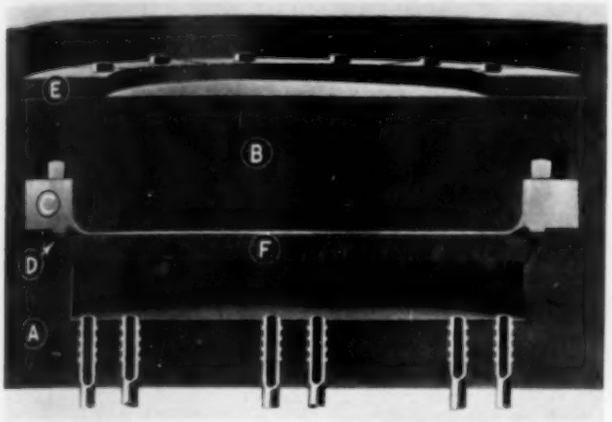
When the labor cost, as well as that for the material, is taken into account, it will be found that the home-made type of lift fitting will cost as much, or more, than the manufactured article.

Current Types of Gaskets and Closures

By Parameter

Usual bolted joint constructions are largely being modified or superseded by more reliable, more easily maintained, and more economical forms. In this—the second of two articles—the author describes several such patented types

THE "lockhead" construction illustrated in Fig. 1 is in regular service on a number of heaters at pressures of 2600 lb per sq in. (Much higher pressures are practical). Several installations are in service where the heads are 10 ft in diameter and the pressures 180 lb per sq in. The basis of the design is to carry the pressure load upon a very heavy head plate held in place by means of shear blocks *E* set into a groove machined into the main steel forging that forms the shell. The pressure load is transferred to the head plate by means of a flexible steel diaphragm *C*, the circumference of which is sealed with a gasket and a ring of bolts.



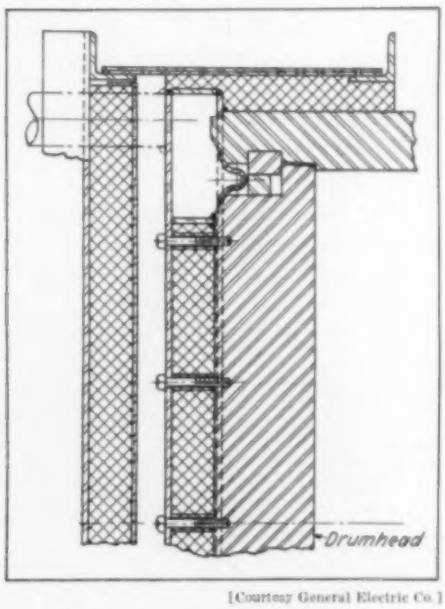
[Courtesy Foster Wheeler Corp.]

Fig. 1—"Lockhead" construction. The plate marked *F* is not an essential part of the design. It has been used in some cases where there might be danger of forming a vacuum in the tubes and thereby reversing the pressure on the diaphragm which would bulge inward due to atmospheric pressure with possible damage. Plate *F*, when used, has holes drilled through it and is merely a support for *C* in the direction of negative pressure

Due to the fact that the pressure load is carried on the head plate, the only load to be carried by the bolting ring is the gasket load and an insignificantly small pressure load. Bolts of approximately 1 in. diameter are ample for this purpose and the construction, therefore, resolves maintenance operations into the work that may be done by ordinary hand wrenches. The only other bolts are small screws to hold the shear blocks from dropping out of place. The design is compact, lends itself to rapid opening up or re-assembly, and no space need be provided in which to manipulate long wrench handles. As the gasketed joint is solely for sealing purposes, it is not affected to any practicable degree by either pressure or temperature variations.

An interesting modification of this design, Fig. 2, was made for the drum of a mercury boiler for sealing in vapors at 140 lb per sq in. and 958 F. The drum head is retained by a circular key cut in sections, which is locked in place by another key. The joint is hermetically sealed by a ring which is welded to the drum and head as shown. During early experiments it was found that there was no packing or method of calking which would resist the mercury vapor and the poisonous fumes are therefore confined by fusion welding every joint in the apparatus.

Fig. 3[1] illustrates a type of cover joint for reaction vessels suitable for either small or large vessels, and said to have been constructed up to one meter in diameter. Internal pressure forces the cover *2* against a ring gasket *1* with a wedge shaped cross-section, which is backed up by a split retainer ring *3* held in a bored recess in the flange of the vessel. The function of assembly ring *4* is to hold the parts in place. The ring gasket should be hard enough, and should be so proportioned, that working stresses will be within the elastic limit and will not cause permanent set of the material. It is claimed that this joint is self sealing and retains its tightness even under fluctuating service pressures. Assembling or dismantling are convenient and easy; the lack of heavy bolting and flanges bring about a considerable reduction in size, with corresponding economy of construction.



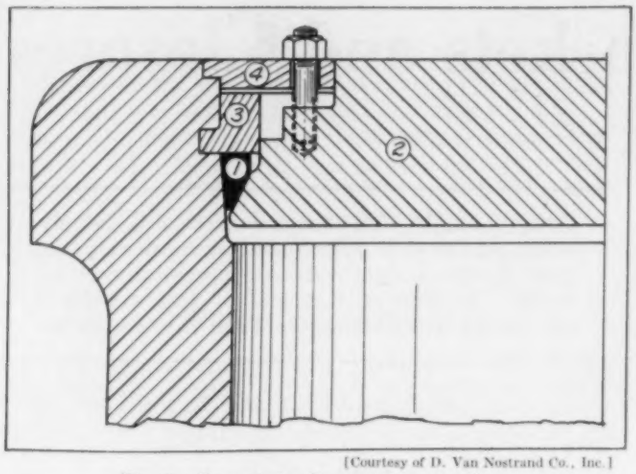
[Courtesy General Electric Co.]

Fig. 2—Drum head attachment for mercury boiler

A closure intended for vessels over 1 ft in diameter, subject to gas pressures of 3000 to 6000 lb per sq in., such as an ammonia converter, has been patented.

The principal feature of this apparatus, Fig. 4, which is partly self sealing, is the steel lens ring gasket *B*. The

[1] See "High Pressure Chemical Plant," by Tongue. (D. Van Nostrand Co., Inc.)



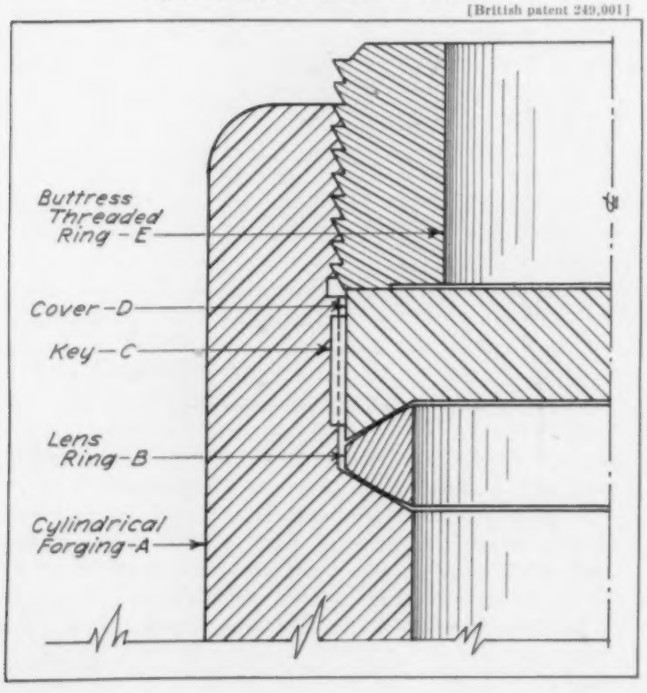
[Courtesy of D. Van Nostrand Co., Inc.]

Fig. 3—Cover joint for reaction vessels

gasket contact surfaces are preferably portions of spheres, with their centers lying on the axis of the ring, thus making a linear contact with the cone shaped seats of the cylinder *A* and cover plate *D*. It is essential that gasket pressure to seal the joint be applied uniformly around the ring, also that the compressive forces be parallel to the axis of the ring and that they pass through the line of contact of the sealing faces. This normal pressure is applied by means of the buttress threaded ring *E* and the cover plate *D;* the latter prevented from rotating by the key *C*. It has been found that angles of from 30 deg to 45 deg for the conical seating surfaces are most effective. Application of pressure to close the joint results in an elastic deformation which forms contact surfaces of a certain width. As a result of internal pressure, the cover plate will tend to assume a spherical shape, and also to lift and thus break the joint, but at the same time, and also as a result of internal pressure, the lens ring will tend to expand and to close the joint.

Fig. 4—Joint for ammonia converter

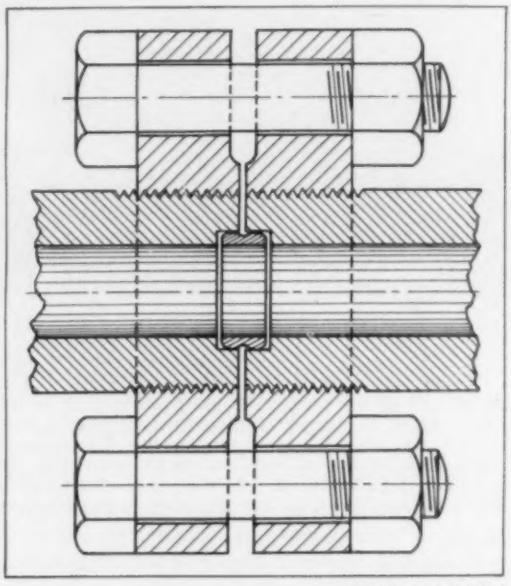
[British patent 249,001]



Since both the internal forces and bolting forces are applied at the line of contact of the joint there is only a slight relative movement of cover and ring, which is taken up by the elastic expansion of the ring. The design is adequate, both for high pressure and for pressure fluctuations.

The "wave ring" joint, Fig. 5, which is truly self-sealing,[2] consists of two equal parallel cylindrical sockets machined in the ends of the members to be joined, these being connected by a hard metal ring with a double convex outer surface, the ring making a force fit with each socket. The sealing pressure is entirely in a radial direction and the bolts do not have to be pulled tight, as they serve only to assemble the joint and to resist the axial force due to fluid thrust.

This joint was developed initially only for high pressure chemical research, and has withstood pressures up to 175,000 lb per sq in. Moreover, at a working pressure of 40,000 lb per sq in. such joints have been opened and remade a hundred times without injury and without



[British patent 397,966]

Fig. 5—Wave ring joint

leakage at any pressure within the working range. They have been used successfully for steam piping up to 12 in. diameter, the total temperature of the steam having been 570 F; also for 1 in. tubing with steam at 3400 lb per sq in. and 734 F.

Since flanges and bolting for this design may be of minimum proportions, considerable saving in weight and cost is effected. However, definite tolerances in ring and socket dimensions are imperative and for service conditions where corrosion is a factor this joint does not appear to be suitable.

The pressure filled gasket of Fig. 6 is really resilient; in addition to this characteristic it resembles the wave ring with respect to its self sealing properties and also

[*Concluded on page* 568]

[2] *Engineering*, 1933, 136, 32; 1936, 141, 192.

Properties of Mixtures of Air and Saturated Water Vapor for Barometric Pressures from 22 In. to 32 In. of Mercury

By William Goodman*

Barometric Pressure 24 In. of Mercury

All tabulated values (except temperatures and pressures) are quantities per pound of dry air in the mixture.

Temp. Deg Fahr	Absolute Humidity		Volume		Humid Specific Heat	1061 w	Enthalpy (Heat Content)		Temp. Deg Fahr	Absolute Humidity		Volume		Humid Specific Heat	1061 w	Enthalpy (Heat Content)	
	Pounds	Grains	Dry Air Cu Ft	Volume Increase of Humid Air Per Cent			Water Vapor Btu	Air-Vapor Mixture Btu		Pounds	Grains	Dry Air Cu Ft	Volume Increase of Humid Air Per Cent			Water Vapor Btu	Air-Vapor Mixture Btu
t	w	w	v_D		s^*		h_v	h	t	w	w	v_D		s^*		h_v	h
1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
−40	0.0001451	1.016	13.19	0.02333	0.2401	0.1539	0.1513	−9.449	40	0.006493	45.45	15.70	1.043	0.2429	6.889	7.007	16.61
−39	.0001536	1.075	13.22	.02469	.2401	.1630	.1603	−9.200	41	.006752	47.26	15.73	1.085	.2430	7.163	7.290	17.13
−38	.0001627	1.139	13.25	.02616	.2401	.1726	.1699	−8.950	42	.007021	49.15	15.77	1.128	.2432	7.449	7.584	17.66
−37	.0001723	1.206	13.28	.02770	.2401	.1828	.1799	−8.700	43	.007298	51.09	15.80	1.173	.2433	7.743	7.886	18.21
−36	.0001824	1.277	13.31	.02932	.2401	.1935	.1905	−8.450	44	.007586	53.10	15.83	1.219	.2434	8.049	8.200	18.76
−35	0.0001928	1.350	13.35	0.03100	0.2401	0.2046	0.2015	−8.199	45	0.007888	55.22	15.86	1.268	0.2436	8.369	8.531	19.33
−34	.0002041	1.428	13.38	.03281	.2401	.2165	.2134	−7.947	46	.008197	57.38	15.89	1.317	.2437	8.697	8.868	19.91
−33	.0002157	1.510	13.41	.03468	.2401	.2289	.2257	−7.694	47	.008516	59.61	15.92	1.368	.2438	9.035	9.217	20.50
−32	.0002281	1.597	13.44	.03668	.2401	.2421	.2388	−7.441	48	.008848	61.94	15.95	1.422	.2440	9.388	9.581	21.10
−31	.0002409	1.686	13.47	.03873	.2401	.2556	.2523	−7.188	49	.009191	64.33	15.99	1.477	.2441	9.751	9.955	21.72
−30	0.0002547	1.783	13.50	0.04095	0.2401	0.2702	0.2668	−6.933	50	0.009548	66.84	16.02	1.534	0.2443	10.13	10.35	22.35
−29	.0002690	1.883	13.53	.04325	.2401	.2854	.2819	−6.678	51	.009919	69.43	16.05	1.594	.2445	10.52	10.75	22.99
−28	.0002838	1.987	13.57	.04563	.2401	.3011	.2976	−6.423	52	.01029	72.05	16.08	1.654	.2446	10.92	11.16	23.64
−27	.0002997	2.098	13.60	.04819	.2401	.3180	.3144	−6.166	53	.01069	74.84	16.11	1.718	.2448	11.34	11.60	24.32
−26	.0003162	2.213	13.63	.05083	.2401	.3355	.3318	−5.908	54	.01109	77.66	16.14	1.783	.2450	11.77	12.04	25.00
−25	0.0003335	2.334	13.66	0.05362	0.2402	0.3538	0.3501	−5.650	55	0.01151	80.60	16.17	1.850	0.2452	12.22	12.50	25.70
−24	.0003518	2.463	13.69	.05656	.2402	.3732	.3695	−5.391	56	.01195	83.65	16.21	1.919	.2454	12.68	12.98	26.42
−23	.0003711	2.597	13.72	.05966	.2402	.3937	.3899	−5.130	57	.01240	86.77	16.24	1.992	.2456	13.15	13.47	27.15
−22	.0003910	2.737	13.75	.06286	.2402	.4148	.4110	−4.869	58	.01286	90.05	16.27	2.066	.2458	13.65	13.98	27.90
−21	.0004120	2.884	13.79	.06623	.2402	.4371	.4332	−4.607	59	.01334	93.37	16.30	2.143	.2460	14.15	14.51	28.67
−20	0.0004341	3.039	13.82	0.06979	0.2402	0.4606	0.4567	−4.343	60	0.01383	96.80	16.33	2.222	0.2462	14.67	15.05	29.45
−19	.0004573	3.201	13.85	.07353	.2402	.4852	.4814	−4.079	61	.01434	100.4	16.36	2.305	.2465	15.22	15.61	30.25
−18	.0004813	3.369	13.88	.07738	.2402	.5107	.5068	−3.813	62	.01488	104.2	16.39	2.390	.2467	15.79	16.20	31.08
−17	.0005066	3.546	13.91	.08144	.2402	.5375	.5336	−3.546	63	.01543	108.0	16.43	2.477	.2469	16.37	16.81	31.93
−16	.0005334	3.734	13.94	.08576	.2402	.5660	.5621	−3.278	64	.01598	111.9	16.46	2.568	.2472	16.96	17.42	32.78
−15	0.0005609	3.926	13.97	0.09017	0.2403	0.5951	0.5913	−3.009	65	0.01658	116.0	16.49	2.661	0.2475	17.59	18.07	33.67
−14	.0005900	4.130	14.01	.09485	.2403	.6260	.6223	−2.738	66	.01717	120.2	16.52	2.758	.2477	18.22	18.73	34.57
−13	.0006206	4.344	14.04	.09978	.2403	.6585	.6549	−2.465	67	.01780	124.6	16.55	2.859	.2480	18.89	19.42	35.50
−12	.0006524	4.567	14.07	.1049	.2403	.6921	.6886	−2.191	68	.01844	129.1	16.58	2.961	.2483	19.56	20.13	36.45
−11	.0006851	4.796	14.10	.1102	.2403	.7269	.7236	−1.916	69	.01910	133.7	16.61	3.067	.2486	20.27	20.86	37.42
−10	0.0007206	5.044	14.13	0.1159	0.2403	0.7645	0.7613	−1.639	70	0.01979	138.5	16.65	3.178	0.2489	21.00	21.61	38.41
−9	.0007565	5.296	14.16	.1216	.2403	.8027	.7997	−1.360	71	.02049	143.4	16.68	3.291	.2492	21.74	22.39	39.43
−8	.0007951	5.566	14.19	.1278	.2404	.8436	.8408	−1.079	72	.02123	148.6	16.71	3.409	.2496	22.52	23.21	40.49
−7	.0008343	5.840	14.23	.1341	.2404	.8852	.8826	−0.7974	73	.02198	153.9	16.74	3.530	.2499	23.32	24.04	41.56
−6	.0008756	6.129	14.26	.1408	.2404	.9290	.9266	−0.5134	74	.02276	159.3	16.77	3.655	.2502	24.15	24.91	42.67
−5	0.0009195	6.436	14.29	0.1478	0.2404	0.9756	0.9736	−0.2265	75	0.02356	164.9	16.80	3.783	0.2506	25.00	25.79	43.79
−4	.0009645	6.751	14.32	.1551	.2404	1.023	1.022	+0.0616	76	.02439	170.7	16.83	3.916	.2510	25.88	26.70	44.94
−3	.001012	7.081	14.35	.1626	.2405	1.073	1.072	0.3520	77	.02526	176.8	16.87	4.056	.2514	26.80	27.67	46.15
−2	.001061	7.429	14.38	.1706	.2405	1.126	1.125	0.6451	78	.02614	183.0	16.90	4.197	.2518	27.74	28.65	47.37
−1	.001113	7.792	14.41	.1790	.2405	1.181	1.181	0.9407	79	.02705	189.4	16.93	4.343	.2522	28.70	29.65	48.61
0	0.001166	8.163	14.45	0.1875	0.2405	1.237	1.237	1.237	80	0.02799	195.9	16.96	4.493	0.2526	29.70	30.69	49.89
1	.001222	8.553	14.48	.1964	.2406	1.296	1.297	1.537	81	.02897	202.8	16.99	4.651	.2530	30.74	31.78	51.22
2	.001280	8.961	14.51	.2058	.2406	1.358	1.359	1.839	82	.02997	209.8	17.02	4.811	.2535	31.80	32.89	52.57
3	.001341	9.388	14.54	.2156	.2406	1.423	1.425	2.145	83	.03101	217.1	17.05	4.977	.2540	32.90	34.04	53.96
4	.001405	9.836	14.57	.2259	.2406	1.491	1.493	2.453	84	.03206	224.4	17.09	5.147	.2544	34.02	35.22	55.38
5	0.001471	10.30	14.60	0.2365	0.2407	1.561	1.564	2.764	85	0.03317	232.2	17.12	5.324	0.2549	35.19	36.44	56.84
6	.001539	10.78	14.63	.2475	.2407	1.633	1.637	3.077	86	.03431	240.2	17.15	5.506	.2554	36.40	37.71	58.35
7	.001611	11.28	14.67	.2590	.2407	1.709	1.714	3.394	87	.03548	248.4	17.18	5.694	.2560	37.64	39.02	59.90
8	.001686	11.80	14.70	.2710	.2408	1.789	1.795	3.715	88	.03668	256.8	17.21	5.888	.2565	38.92	40.36	61.48
9	.001764	12.35	14.73	.2836	.2408	1.872	1.879	4.039	89	.03795	265.6	17.24	6.090	.2571	40.26	41.76	63.12
10	0.001844	12.91	14.76	0.2965	0.2408	1.957	1.965	4.365	90	0.03923	274.6	17.27	6.295	0.2577	41.63	43.19	64.79
11	.001930	13.51	14.79	.3103	.2409	2.048	2.058	4.698	91	.04057	284.0	17.31	6.510	.2583	43.04	44.68	66.52
12	.002018	14.13	14.82	.3244	.2409	2.141	2.152	5.032	92	.04192	293.5	17.34	6.728	.2589	44.48	46.19	68.27
13	.002107	14.75	14.85	.3388	.2410	2.236	2.248	5.368	93	.04335	303.4	17.37	6.955	.2595	45.99	47.78	70.10
14	.002205	15.43	14.89	.3544	.2410	2.339	2.353	5.713	94	.04482	313.7	17.40	7.189	.2602	47.55	49.41	71.97
15	0.002303	16.12	14.92	0.3703	0.2410	2.444	2.460	6.060	95	0.04632	324.2	17.43	7.431	0.2608	49.14	51.09	73.89
16	.002405	16.84	14.95	.3867	.2411	2.552	2.569	6.409	96	.04786	335.1	17.46	7.679	.2615	50.78	52.82	75.86
17	.002512	17.58	14.98	.4038	.2411	2.665	2.684	6.764	97	.04946	346.2	17.49	7.935	.2623	52.47	54.59	77.87
18	.002623	18.36	15.01	.4217	.2412	2.783	2.804	7.124	98	.05113	357.9	17.53	8.202	.2630	54.25	56.47	79.99
19	.002739	19.17	15.04	.4404	.2412	2.906	2.930	7.490	99	.05284	369.9	17.56	8.477	.2638	56.06	58.38	82.14
20	0.002858	20.00	15.07	0.4594	0.2413	3.032	3.058	7.858	100	0.05459	382.1	17.59	8.756	0.2646	57.92	60.33	84.33
21	.002984	20.89	15.11	.4797	.2413	3.166	3.194	8.234	101	.05641	394.8	17.62	9.048	.2654	59.85	62.36	86.60
22	.003112	21.79	15.14	.5004	.2414	3.302	3.333	8.613	102	.05829	408.0	17.65	9.348	.2662	61.84	64.47	88.95
23	.003247	22.73	15.17	.5220	.2415	3.445	3.479	8.999	103	.06023	421.6	17.68	9.657	.2671	63.90	66.64	91.36
24	.003387	23.71	15.20	.5445	.2415	3.594	3.630	9.390	104	.06223	435.6	17.71	9.980	.2680	66.02	68.88	93.84
25	0.003530	24.71	15.23	0.5675	0.2416	3.745	3.785	9.785	105	0.06427	449.9	17.75	10.31	0.2689	68.20	71.17	96.37
26	.003683	25.78	15.26	.5921	.2417	3.907	3.950	10.19	106	.06640	464.8	17.78	10.65	.2699	70.46	73.56	99.00
27	.003838	26.87	15.29	.6170	.2417	4.072	4.119	10.60	107	.06862	480.3	17.81	11.00	.2709	72.80	76.04	101.72
28	.003999	27.99	15.33	.6429	.2418	4.243	4.293	11.01	108	.07088	496.1	17.84	11.37	.2719	75.20	78.57	104.49
29	.004167	29.17	15.36	.6699	.2419	4.421	4.476	11.44	109	.07322	512.6	17.87	11.74	.2730	77.69	81.20	107.36
30	0.004342	30.40	15.39	0.6981	0.2420	4.607	4.666	11.87	110	0.07565	529.5	17.90	12.13	0.2740	80.26	83.93	110.33
31	.004525	31.68	15.42	.7275	.2420	4.801	4.865	12.31	111	.07812	546.8	17.93	12.53	.2752	82.88	86.70	113.34
32	.004709	32.96	15.45	.7569	.2421	4.996	5.066	12.75	112	.08071	564.9	17.97	12.94	.2763	85.63	89.61	116.49
33	.004908	34.35	15.48	.7887	.2422	5.207	5.282	13.20	113	.08338	583.7	18.00	13.37	.2775	88.46	92.61	119.73
34	.005111	35.77	15.51	.8214	.2423	5.422	5.503	13.66	114	.08613	602.9	18.03	13.80	.2788	91.38	95.70	123.06
35	0.005319	37.24	15.55	0.8551	0.2424	5.644	5.730	14.13	115	0.08896	622.7	18.06	14.25	0.2800	94.38	98.89	126.49
36	.005539	38.77	15.58	0.8903	.2425	5.877	5.969	14.61	116	.09192	643.4	18.09	14.73	.2814	97.52	102.21	130.05
37	.005766	40.36	15.61	0.9264	.2426	6.117	6.215	15.10	117	.09495	664.7	18.12	15.21	.2827	100.74	105.62	133.70
38	.006000	42.00	15.64	0.9642	.2427	6.366	6.470	15.59	118	.09807	686.5	18.15	15.71	.2841	104.05	109.13	137.45
39	.006242	43.69	15.67	1.003	.2428	6.622	6.734	16.09	119	.1013	708.9	18.19	16.23	.2856	107.45	112.73	141.29

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(Tabulations for 22 in. and 23 in. were published in July and August.)

Properties of Mixtures of Air and Saturated Water Vapor—(Continued)

Barometric Pressure 24 In. of Mercury

All tabulated values (except temperatures and pressures) are quantities per pound of dry air in the mixture.

Temp. Deg Fahr	Absolute Humidity		Volume		Humid Specific Heat	1061w	Enthalpy (Heat Content)		Temp. Deg Fahr	Absolute Humidity		Volume		Humid Specific Heat	1061w	Enthalpy (Heat Content)	
	Pounds	Grains	Dry Air Cu Ft	Volume Increase of Humid Air Per Cent			Water Vapor Btu	Air-Vapor Mixture Btu		Pounds	Grains	Dry Air Cu Ft	Volume Increase of Humid Air Per Cent			Water Vapor Btu	Air-Vapor Mixture Btu
t	w	w	v_D		s''		h_V	h		w	w	v_D		s''		h_V	h
1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
120	0.1047	732.6	18.22	16.77	0.2871	111.03	116.55	145.35	160	0.4214	2949.5	19.47	67.26	0.4296	447.06	476.22	514.62
121	.1081	756.8	18.25	17.32	.2887	114.70	120.45	149.49	161	.4386	3070.5	19.51	70.01	.4374	465.40	495.93	534.57
122	.1116	781.5	18.28	17.89	.2902	118.45	124.42	153.70	162	.4570	3198.8	19.54	72.93	.4456	484.85	516.83	555.71
123	.1154	807.6	18.31	18.48	.2919	122.41	128.63	158.15	163	.4762	3333.4	19.57	75.98	.4543	505.25	538.77	577.89
124	.1192	834.3	18.34	19.09	.2936	126.45	132.92	162.68	164	.4967	3476.6	19.60	79.25	.4635	526.95	562.11	601.47
125	0.1231	862.0	18.37	19.72	0.2954	130.65	137.40	167.40	165	0.5182	3627.5	19.63	82.67	0.4732	549.82	586.72	626.32
126	.1272	890.7	18.41	20.38	.2973	135.00	142.03	172.27	166	.5412	3788.7	19.66	86.34	.4836	574.26	613.01	652.85
127	.1315	920.4	18.44	21.05	.2992	139.50	146.81	177.29	167	.5655	3958.4	19.69	90.19	.4945	599.97	640.69	680.77
128	.1359	951.2	18.47	21.76	.3012	144.17	151.78	182.50	168	.5915	4140.7	19.73	94.32	.5062	627.61	670.44	710.76
129	.1404	982.9	18.50	22.49	.3032	148.99	156.91	187.87	169	.6190	4332.8	19.76	98.68	.5185	656.73	701.79	742.35
130	0.1452	1016.5	18.53	23.24	0.3053	154.07	162.33	193.53	170	0.6485	4539.3	19.79	103.4	0.5318	688.03	735.50	776.30
131	.1501	1050.6	18.56	24.02	.3075	159.25	167.85	199.29	171	.6798	4758.7	19.82	108.4	.5459	721.28	771.31	812.35
132	.1550	1085.1	18.59	24.82	.3098	164.47	173.41	205.09	172	.7136	4995.3	19.85	113.7	.5611	757.14	809.95	851.23
133	.1603	1122.0	18.63	25.66	.3121	170.07	179.38	211.30	173	.7501	5250.8	19.88	119.5	.5776	795.87	851.68	893.20
134	.1658	1160.3	18.66	26.52	.3146	175.87	185.57	217.73	174	.7892	5524.1	19.91	125.7	.5951	837.29	896.32	938.08
135	0.1714	1199.5	18.69	27.43	0.3171	181.81	191.91	224.31	175	0.8308	5815.3	19.94	132.3	0.6138	881.44	943.91	985.91
136	.1772	1240.2	18.72	28.35	.3197	187.98	198.48	231.12	176	0.8762	6133.3	19.98	139.6	.6343	929.63	995.87	1038.1
137	.1833	1282.8	18.75	29.32	.3225	194.44	205.40	238.28	177	0.9257	6479.8	20.01	147.4	.6566	982.15	1052.5	1095.0
138	.1897	1327.6	18.78	30.34	.3254	201.23	212.65	245.77	178	0.9796	6857.3	20.04	156.0	.6808	1039.4	1114.2	1156.9
139	.1961	1373.0	18.81	31.37	.3283	208.10	219.99	253.35	179	1.038	7268.8	20.07	165.3	.7073	1101.7	1181.4	1224.4
140	0.2029	1420.5	18.85	32.46	0.3313	215.31	227.69	261.29	180	1.103	7719.6	20.10	175.6	0.7363	1170.1	1255.1	1298.3
141	.2099	1469.4	18.88	33.58	.3345	222.71	235.60	269.44	181	1.174	8216.6	20.13	186.9	.7682	1245.4	1336.4	1379.8
142	.2174	1521.8	18.91	34.76	.3378	230.66	244.10	278.18	182	1.253	8768.2	20.16	199.3	.8037	1329.0	1426.6	1470.3
143	.2249	1574.4	18.94	35.98	.3412	238.64	252.63	286.95	183	1.340	9378.6	20.20	213.2	.8429	1421.5	1526.4	1570.4
144	.2328	1629.9	18.97	37.24	.3448	247.04	261.62	296.18	184	1.438	10065	20.23	228.7	.8870	1525.5	1638.7	1682.8
145	0.2411	1687.9	19.00	38.57	0.3485	255.84	271.05	305.85	185	1.548	10,835	20.26	246.2	0.9366	1642.3	1764.8	1809.2
146	.2499	1749.1	19.03	39.94	.3524	265.11	280.98	316.02	186	1.673	11,709	20.29	266.0	0.9927	1774.7	1907.7	1952.4
147	.2589	1812.0	19.07	41.37	.3565	274.65	291.19	326.47	187	1.815	12,706	20.32	288.7	1.057	1925.9	2071.0	2115.8
148	.2682	1877.7	19.10	42.87	.3607	284.60	301.85	337.37	188	1.980	13,859	20.35	314.8	1.131	2100.7	2259.7	2304.8
149	.2780	1945.9	19.13	44.43	.3651	294.94	312.92	348.68	189	2.172	15,207	20.38	345.4	1.218	2304.9	2480.2	2525.6
150	0.2883	2018.2	19.16	46.07	0.3697	305.90	324.67	360.67	190	2.399	16,795	20.42	381.3	1.320	2545.7	2740.0	2785.6
151	.2990	2093.0	19.19	47.78	.3746	317.24	336.82	373.06	191	2.672	18,702	20.45	424.6	1.442	2834.7	3052.2	3098.0
152	.3098	2168.7	19.22	49.55	.3794	328.72	349.14	385.62	192	3.004	21,027	20.48	477.3	1.592	3187.1	3432.9	3478.9
153	.3219	2253.2	19.25	51.42	.3849	341.51	362.86	399.58	193	3.420	23,938	20.51	543.3	1.779	3628.3	3909.4	3955.7
154	.3341	2338.7	19.29	53.36	.3904	354.48	376.77	413.73	194	3.953	27,669	20.54	627.9	2.019	4193.8	4520.3	4566.9
155	0.3470	2429.2	19.32	55.42	0.3962	368.20	391.49	428.69	195	4.661	32,628	20.57	740.3	2.338	4945.4	5332.3	5379.1
156	.3601	2521.0	19.35	57.51	.4021	382.11	406.45	443.89	196	5.649	39,540	20.60	897.1	2.782	5993.1	6464.1	6511.2
157	.3743	2620.3	19.38	59.79	.4085	397.16	422.62	460.30	197	7.123	49,858	20.64	1130.8	3.445	7557.1	8153.2	8200.5
158	.3894	2725.5	19.41	62.16	.4152	413.11	439.74	477.66	198	9.552	66,861	20.67	1516.2	4.538	10134	10937	10985
159	.4049	2834.3	19.44	64.64	.4222	429.60	457.46	495.62	199	12.82	89,719	20.70	2034.3	6.008	13599	14682	14730
									200	27.75	194,243	20.73	4402.9	12.73	29442	31798	31846

Gaskets and Closures

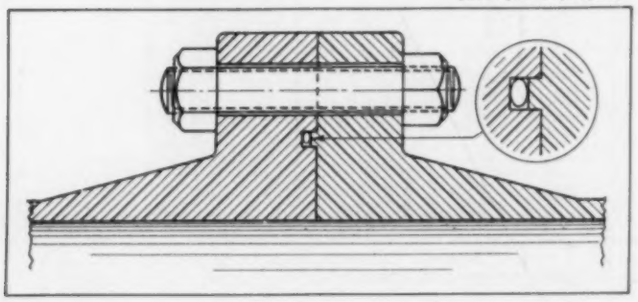
[*Concluded from page 566*]

with respect to requiring a minimum of bolt tension to draw the parts of the joint together. It is a hermetically sealed tube of copper, cupro-nickel, monel or other suitable metal, formed into circular, oval or other specified shape; the tube being loaded internally, before installation, with a non-explosive gas under pressure, which, it is claimed, will not liquefy at any temperature above —295 F.

Working pressures up to 20,000 lb per sq in. at room temperature have been held by these gaskets for long periods of time, and it is said they have been used under actual working conditions of 1800 lb per sq in. and 1000 F. The usual construction calls for $\frac{1}{8}$ in., $\frac{3}{16}$ in., or $\frac{1}{4}$ in. tubing, with the internal pressure set to provide a "recovery" of 0.01 in. after compression of the gasket. Ordinarily, for pressures up to 2000 lb per sq in., these gaskets are compressed one-third of their cross-section; they are pulled down to as much as one-half of their original diameter for higher pressures. Grooves should be flat bottomed and at least $\frac{1}{32}$ in. wider than the compressed gasket, as indicated in Fig. 6. For high pressure work the ring should make definite contact with the outer wall of the groove.

Fig. 6—Flange joint with pressure filled gasket

[U. S. patent 1,917,175]



These gaskets have been made in sizes varying from less than 1 in. up to 6 ft in diameter and have been applied to a very wide variety of uses. Although better known abroad than here, they are protected by an American patent and are now manufactured and marketed in the United States.

Mechanically joined pipe lines or parts of pressure apparatus have frequently been the least satisfactory details of the equipment of which they are such essential components. The foregoing information concerning outstanding designs has been presented with the purpose of assisting engineers and of stimulating further research and progress in joint design.

[*This is the second of two articles on gaskets and closures.*]

Heating · Piping · Air Conditioning

AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

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All who hold Society membership are eligible to join the nearest local Chapter. Local monthly meetings are usually held from October to May in 28 cities of the United States and Canada. The fall season's program will commence soon.

• • •

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That over 10,000 copies of The Guide 1939 are now being used . . . that 18 research projects are being sponsored by the Society . . . that 300 new members have applied for membership for this year.

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Effect of Size and Type of Air Inlet and Outlet on the Heat Output of Convectors

By A. P. Kratz*, M. K. Fahnestock**, and E. L. Broderick***, (*MEMBERS*), Urbana, Ill.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the University of Illinois.

THE object of this investigation was to determine the effect of different sizes and types of inlets and outlets on the heat output of convectors, and, if possible, to correlate the areas of such inlets and outlets with the free area through the heating unit. All tests were run in a warm wall test booth and in every case the distance between the back of the convector cabinet and the wall was 2½ in. For the principal series of tests three convectors were used, each consisting of a different type of heating unit fitted into a special cabinet which was similar to a commercial cabinet except that it was provided with a means for adjusting the widths of inlets and outlets. In all cases a grilled front outlet was used in connection with either a grilled or an ungrilled front inlet, and a program of tests was conducted to determine the performance of the convectors with different combinations of widths of inlets and outlets. Additional tests were run in order to determine the effect of reducing the length of the inlet relative to the length of the cabinet, of introducing different arrangements of vanes in connection with the inlets and outlets, of using an inlet at both the front and back of the cabinet, and of locating the outlet in the top of the cabinet. Some tests were also run on a special convector having a steam heated humidifying pan in the upper part of the cabinet.

Description of Apparatus

The warm wall test booth was constructed in accordance with the specifications given in the ASHVE Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code)[1] and has been described in previous papers[2, 3, 4]. An elevation section of the test booth and a detail of the piping are shown in Figs. 1 and 2. For most of the tests the piping used differed slightly from that shown in the Code, in that the by-pass for determining the piping correction was omitted. In some of the earlier tests this by-pass was used, but it later proved to be unnecessary. For all later tests the piping correction was found to be constant, and was obtained by means of separate tests, for the purpose of which a length of heavily insulated pipe was substituted for the convector. As compared with the apparatus specified in the Code, a more accurate determination of the water weights was made possible by using a receiver having a section of ¾-in. pipe instead of 2-in. pipe placed opposite the section of the gage glass used for reading the condensate level. The steam pressure was more accurately obtained by using a water manometer instead of one filled with mercury.

The three types of heating units used for most of the tests are shown in Fig. 3 together with one of the special cabinets. The heating unit for Convector No. 35 consisted of finned cast-iron elements bolted together, and had a nominal height of 6 in. The heating unit for Convector No. 36 consisted of three round copper tubes and for Convector No. 37 of four oval copper tubes each having attached fins and extending between cast-iron headers. The height of the heating unit for Convector No. 36 was 2 in., and for Convector No. 37 was 5¼ in.

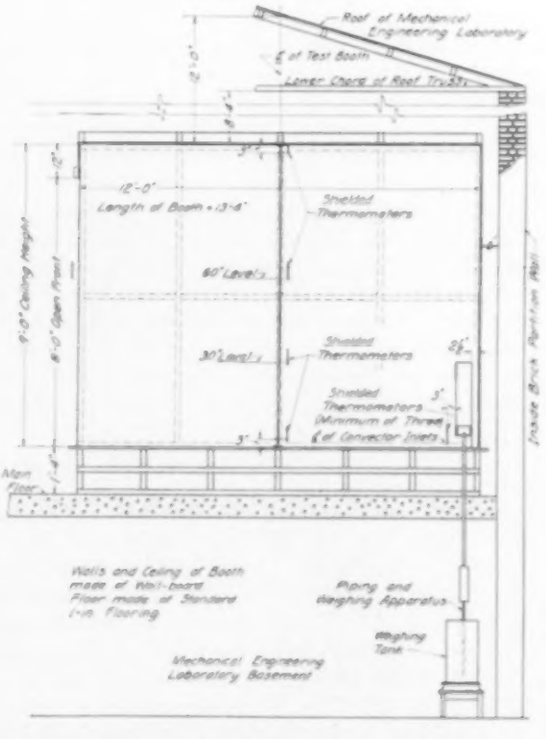

Fig. 1—Section of warm wall test booth

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[1]ASHVE Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code). (ASHVE TRANSACTIONS, Vol. 37, 1931, p. 367. Revised January 1936, ASHVE TRANSACTIONS, Vol. 42, 1936, p. 29.)

[2]ASHVE Research Report No. 938—Tests of Convectors in a Warm Wall Testing Booth, by A. P. Kratz, M. K. Fahnestock, and E. L. Broderick. (ASHVE TRANSACTIONS, Vol. 38, 1932, p. 511.)

[3]ASHVE Research Report No. 963—Tests of Convectors in a Warm Wall Testing Booth, Part II, by A. P. Kratz, M. K. Fahnestock, and E. L. Broderick. (ASHVE TRANSACTIONS, Vol. 39, 1933, p. 319.)

[4]ASHVE Research Report No. 998—Factors Affecting the Heat Output of Convectors, by A. P. Kratz, M. K. Fahnestock, and E. L. Broderick. (ASHVE TRANSACTIONS, Vol. 40, 1934, p. 443.)

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Mackinac Island, Mich., July, 1939.

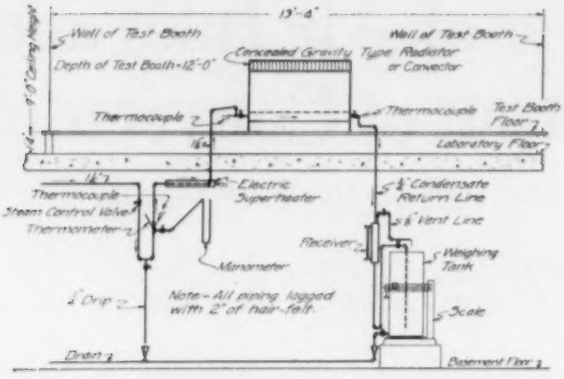


Fig. 2—Detail of piping for warm wall test booth

The principal dimensions of the three convectors are given in Fig. 4. The cabinets for these convectors were of the same height and, since the heating units were of approximately the same length, the lengths of the inlets and outlets were practically the same. The area of the inlet or outlet was varied by maintaining the length constant and changing the width. In the case of the inlet a plate partly covered the opening as shown in Fig. 3 and the width of the opening was increased by raising the plate. The maximum width was obtained when the lower edge of the plate was on a level with the bottom of the heating unit. A similar plate with a serrated upper edge partly covered the outlet and was lowered in order to increase the width of the opening. The type of grilles employed, as shown in Fig. 3, had ¾-in. slots and ¼-in. frets. The widths of inlets and outlets used in the test program are given in Fig. 4.

Details of the special arrangements used in connection with tests not included in the principal test series are shown in the insets appearing in conjunction with the different curves representing the test results. (See Figs. 10, 11, and 12.)

Modifications of Convector No. 35 were used for all of the tests run with a reduced length of inlet and with different arrangements of vanes at inlets and outlets. (See Fig. 10.) Cardboard cut to the proper shape and size was used to block the ends of the inlets and sheet metal was used for the different vanes which were fastened in place with metal screws. A vane used at the top of the outlet was formed with a radius equal to the inside width of the cabinet, and one-half of the width was employed for the radii of intermediate vanes used at the center of the inlet and outlet. In one case a special vane was placed just below the outlet and was arranged to form a venturi throat. This vane was 1½ in. wide and reduced the width of the cabinet at that point from 5¾ in. to 4¼ in.

Two convectors were used for the tests on the arrangements having outlets in the top of the cabinets. For the tests conducted with modifications of Convector No. 35 (See Fig. 11) the front outlet was covered, and a grilled outlet similar to the front outlet was provided in the top of the cabinet. This convector had a heating unit 6 in. in height. Convector No. 33, which was used for a second series of tests on a convector having a top outlet, had a cast-iron heating unit 12 in. in height. Convector No. 31 was used for a series of tests on a cabinet having both a front and back inlet. The heating unit was made of cast-iron and was 4½ in. high. The cabinet had a conventional inlet in the front below the heating unit, and a corresponding inlet in the back. One series of tests was run with both inlets open, and one series with the back inlet closed.

Convector No. 11*a* was a modification of Convector No. 11. The heating unit, used in connection with a commercial cabinet, was the same heating unit as that used for Convector No. 37. In Convector No. 11*a* a water pan, or humidifier, was used at the top of the cabinet, and the outlet was made correspondingly wider. This water pan had a false bottom which formed a steam space that was connected by pipes at both ends to the steam headers of the heating unit. (See Fig. 12.) The water in the pan was thus evaporated with heat furnished by the condensation of steam in the space between bottoms. Details of this arrangement are given in the insert shown in connection with the curves representing the results.

Test Procedure

All tests were run with a steam temperature of 216.5 F in the convectors. The relation between the heat output, or the heat equivalent of the net steam condensed, and the temperature of the air at the inlet was established by running tests at different inlet air temperatures varying over a range of from 60 to 80 F. In order to obtain this range of inlet air temperatures, the large laboratory in which the test booth was located was heated or cooled so that the desired inlet air temperature resulted when equilibrium had been attained.

No test observations were made until conditions had remained constant for several hours, as indicated by readings of all thermocouples and thermometers. The condensate was then weighed over the period of one hour, and no test was accepted if the condensate showed more than 2½ per cent deviation in the successive 10-min increments of weight. The total condensate for each test was corrected by subtracting the condensation due to the piping alone as determined by separate tests.

The tests on each of the three principal convectors were divided into four series as indicated in Fig. 4. For Series I the width of the grilled outlet of the cabinet was set at 5 in., and ungrilled inlets of different widths were used. The setting of the grilled outlet remained the same for Series II, but a grilled inlet was used in each case instead of an ungrilled inlet. In Series III and IV the width of the inlet remained constant and different widths of grilled outlets were used. For Series III an

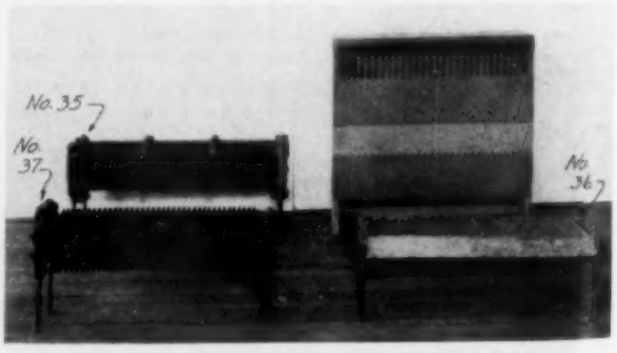


Fig. 3—Convector cabinet and three types of heating units

ungrilled inlet was used, and the width of this inlet was the same as that of the grilled inlet used in Series IV. The width of the grilled inlet selected for Series IV was the same as that of the grilled inlet which gave the best performance in Series II.

No change was made in procedure for any of the other tests with the exception of the tests on Convector No. 11*a*, which had the steam heated water pan in the upper part of the cabinet. During all of the tests on this convector the liquid level in the water pan was observed by means of a gage glass connected to the water pan, and the temperature of the water, or the temperature at which evaporation occurred, was read by means of a thermometer inserted through the top of the cabinet. The weight of water evaporated was obtained from the readings of the gage glass, which was calibrated in terms of the weight of water in the pan.

Results of Tests

General Results

The effect of the widths of the inlets and outlets on the performance of convectors was determined by comparing the heat outputs for the various arrangements. This heat output was expressed in terms of the heat equivalent of the net steam condensed under standard conditions. The latter, as defined by the ASHVE Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code)[5], involves steam in the convector at 215 F and air entering the inlet at 65 F, or a standard temperature difference of 150 F. Since it is not always possible to run a test with a given arrangement under exactly standard conditions, one of two methods may be used to correct the heat output obtained under test conditions to the equivalent heat output with the standard temperature difference of 150 F. The first method consists of running a test under as nearly standard conditions as possible, and then correcting the heat equivalent of the steam condensed, making use of the formula:

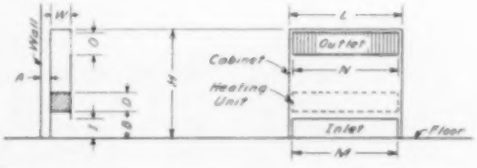
$$H_s = H_t \left[\frac{150}{t_s - t_i} \right]^n \qquad (1)$$

in which H_s = the heat output under standard conditions, Btu per hour.

H_t = the heat output under actual test conditions, Btu per hour.

t_s = temperature of saturated steam under test conditions, degree Fahrenheit.

t_i = average temperature of inlet air under test conditions, degree Fahrenheit.



Convector Number	Type of Heating Unit	Dimensions in Inches A	W	H	L	O	B	M	N	Heating Surface in Sq Ft Primary	Secondary	Ratio*
35	Cast Iron	2 1/2	5 1/8	29	34 1/2	6	7	32 1/2	31 1/8	5.9	19.4	3.3
36	Non-Ferrous	2 1/2	5 1/8	29	30	2	6 1/2	27 1/8	27 1/8	1.7	22.4	13.2
37	Non-Ferrous	2 1/2	5 1/8	29	29 1/2	5 1/2	7	27	26 1/8	3.5	29.2	8.3

*Ratio of Secondary to Primary Heating Surface

Convector Number	Series Number	Inlet Width in Inches = I, Free Area in Sq In. = F	I	F	Outlet Width in Inches = O, Free Area in Sq In. = F	O	F
35	I	Ungrilled	1 2 3 4 5 6 7	64.5 96.8 129.0 161.3 193.5 225.8	Grilled	5	116.1
	II	Grilled	1 2 3 4 5 6	44.1 68.1 92.1 116.1 140.1	Grilled	5	116.1
	III	Ungrilled	6	193.5	Grilled	2 3 4 5 6 7 8	44.1 68.1 92.1 116.1 140.1 164.1 188.1
	IV	Grilled	6	140.1	Grilled	2 3 4 5 6 7 8	44.1 68.1 92.1 116.1 140.1 164.1 188.1
36	I	Ungrilled	3 4 5 6 6 1/2	83.6 111.5 139.4 167.3 188.2	Grilled	5	101.6
	II	Grilled	3 4 5 5 1/2	59.6 80.6 101.6 116.1	Grilled	5	101.6
	III	Ungrilled	5 1/2	160.3	Grilled	3 4 5 6 7	59.6 80.6 101.6 122.6 143.6
	IV	Grilled	5 1/2	117.4	Grilled	3 4 5 6 7	59.6 80.6 101.6 122.6 143.6
37	I	Ungrilled	3 4 5 6 7	81.0 108.0 135.0 162.0 189.0	Grilled	5	98.0
	II	Grilled	3 4 5 6	57.5 77.7 98.0 118.2	Grilled	5	98.0
	III	Ungrilled	6	162.0	Grilled	3 4 5 6 7	57.5 77.7 98.0 118.2 138.5
	IV	Grilled	6	118.2	Grilled	3 4 5 6 7	57.5 77.7 98.0 118.2 138.5

Fig. 4—Dimensions of principal heating units tested

n = an exponent depending on the type of convector (average value of 1.5 recommended in the Code.)

This method involves either determining by tests the value of n for the particular convector, or accepting the average value of 1.5 recommended in the revised Code as being sufficiently accurate for all practical purposes.

A second method consists of running a number of tests at different inlet air temperatures, and thus establishing a performance curve giving the heat outputs over a fairly

[5] Loc. Cit. See Note 1.

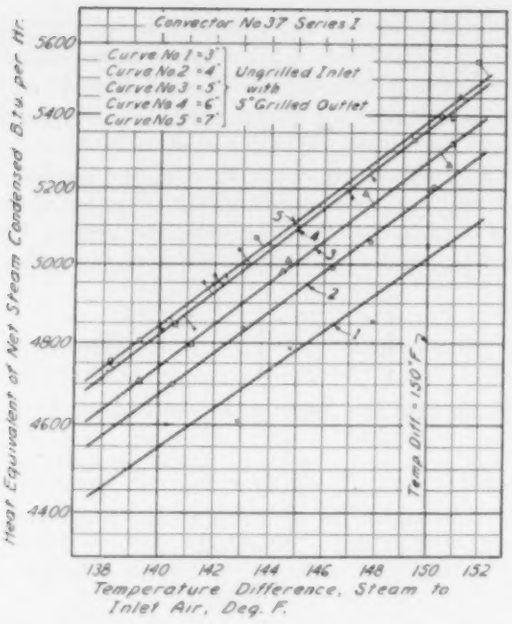

Convector No 37 Series I
Curve No 1 = 3"
Curve No 2 = 4"
Curve No 3 = 5"
Curve No 4 = 6"
Curve No 5 = 7"
Ungrilled Inlet with 5" Grilled Outlet
Temp Diff = 150°F
Heat Equivalent of Net Steam Condensed B.t.u. per Hr.
5600
5400
5200
5000
4800
4600
4400
138 140 142 144 146 148 150 152
Temperature Difference, Steam to Inlet Air, Deg. F.

Fig. 5—Typical performance curves for one series of tests

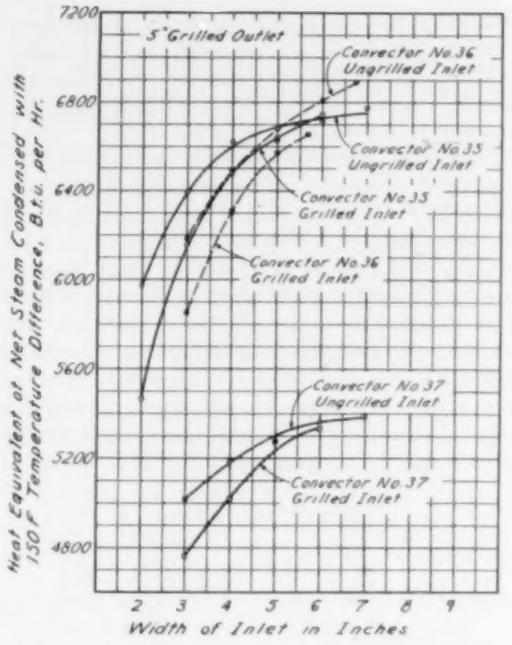


Fig. 6—Heat output of convectors with different widths of inlets

wide range of temperature differences between steam and inlet air. The heat output with the standard temperature difference of 150 F can then be read directly from the curve. The second method was adopted for the purpose of this investigation, and a typical set of performance curves for Convector No. 37 is shown in Fig. 5. In this series of tests Convector No. 37 had a 5-in. grilled outlet which was used in connection with ungrilled inlets varying in width by 1-in. increments from a width of 3 in. to one of 7 in. Each curve represents the results obtained with one width of inlet.

The test data shown in Fig. 5 were plotted to logarithmic scales, and hence the slopes of the resulting straight lines represent corresponding values of n in Equation (1). The values of n obtained from all of the test data resulting from this investigation varied between the limits of 1.43 and 1.69, and the average value agreed reasonably well with that of 1.5 recommended in the ASHVE Code.

Effect of Width of Inlet

A set of performance curves similar to that shown in Fig. 5 was developed for each test series on each of the three convectors listed in Fig. 4. For any given series of tests on each convector the heat output at the standard temperature difference of 150 F was read from one of these curves and plotted against the corresponding width of inlet or outlet. The heat outputs obtained for the three convectors with different widths of inlets, both grilled and ungrilled, are shown in Fig. 6. These curves present the results from Series I run with ungrilled inlets and Series II run with grilled inlets, and each point represents a heat output corresponding to a 150 F temperature difference as read from a curve similar to those shown in Fig. 5.

The curves in Fig. 6 show distinctly that the heat output of each convector was improved as the width of the inlet was increased. The heat output did not attain a definite maximum, but continued to increase until the width of the inlet became equal to the distance between the bottom of the heating unit and the floor. This distance constituted the physical limit beyond which it was not possible to increase the width of the inlet.

In every case, with inlets less than 5½ in. in width, the heat output obtained with the ungrilled inlet was greater than that obtained with the grilled inlet. Furthermore, when the width was increased beyond 5½ in., the heat output obtained with the grilled inlet approached that obtained with the ungrilled inlet, and in the case of Convector No. 35 the two heat outputs became equal with a width of inlet of approximately 5¾ in. It may also be observed that in the case of the ungrilled inlets used with Convectors Nos. 35 and 37 the increase in heat output was not very marked after a width of approximately 5 in. had been reached, while in the case of the grilled inlets a marked increase occurred even up to the attainable limit in width.

Undoubtedly the heat output is affected by the frictional resistance of the convector as a whole, and the best width of inlet is probably determined by the relation between the resistance of the inlet itself and that of the heating unit, since the latter resistance is usually large as compared with that of the cabinet and outlet. The resistance of the inlet is influenced by the free area, and that of the heating unit by both the free area and the height of the unit. With equal widths of inlets the free areas offered by the ungrilled were greater than those offered by the corresponding grilled inlets, and hence resulted in comparatively greater heat outputs. As the width of an ungrilled inlet increased beyond 5 in., however, the decrements in resistance became smaller as compared with the resistance of the heating unit, and the latter became the controlling factor. The effect of the resistance of the heating unit was further accentuated by the fact that any tendency for increased air velocity was immediately reflected in increased resistance of the heating unit, thus tending to offset any

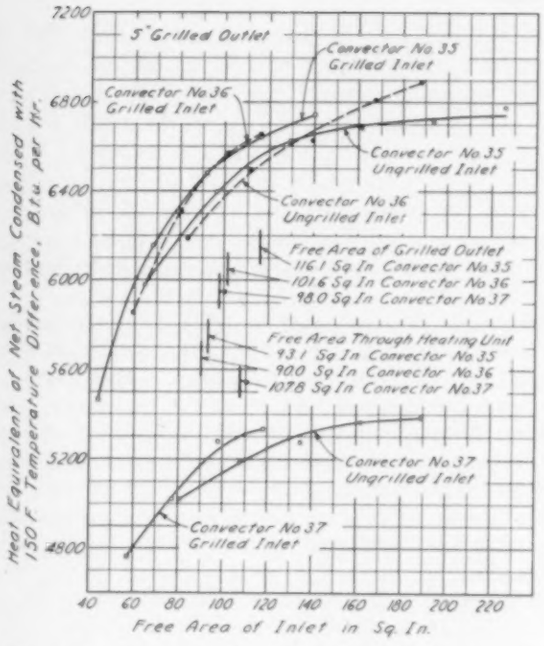

7200
6800
6400
6000
5600
5200
4800
5" Grilled Outlet
Convector No 35 Grilled Inlet
Convector No 36 Grilled Inlet
Convector No 35 Ungrilled Inlet
Convector No 36 Ungrilled Inlet
Free Area of Grilled Outlet
116.1 Sq In Convector No 35
101.6 Sq In Convector No 36
98.0 Sq In Convector No 37
Free Area Through Heating Unit
93.1 Sq In Convector No 35
90.0 Sq In Convector No 36
107.8 Sq In Convector No 37
Convector No 37 Ungrilled Inlet
Convector No 37 Grilled Inlet
Heat Equivalent of Net Steam Condensed with 150 F Temperature Difference, B.t.u. per Hr.
40 60 80 100 120 140 160 180 200 220
Free Area of Inlet in Sq. In.

Fig. 7—Heat output of convectors with inlets having different free areas

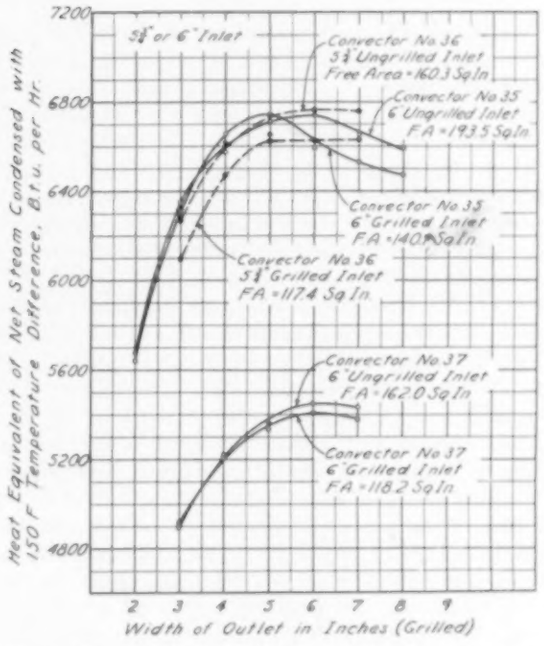

7200
6800
6400
6000
5600
5200
4800
5¾" or 6" Inlet
Convector No 36 5¾ Ungrilled Inlet Free Area = 160.3 Sq In
Convector No 35 6 Ungrilled Inlet F.A. = 193.5 Sq In.
Convector No 35 6" Grilled Inlet F.A. = 140.0 Sq In.
Convector No 36 5¾ Grilled Inlet F.A. = 117.4 Sq In.
Convector No 37 6" Ungrilled Inlet F.A. = 162.0 Sq In.
Convector No 37 6" Grilled Inlet F.A. = 118.2 Sq In.
Heat Equivalent of Net Steam Condensed with 150 F Temperature Difference, B.t.u. per Hr.
2 3 4 5 6 7 8 9
Width of Outlet in Inches (Grilled)

Fig. 8—Heat output of convectors with different widths of outlets

gain resulting from further increase in the width of the inlet. The influence of the height of the heating unit is shown by the fact that, with the ungrilled inlets, in the cases of Convectors Nos. 35 and 37, which had 6-in. and 5¼-in. heating units respectively, a marked decrease in the increments of heat output with the increments of width occurred with widths greater than 5 in., while in the case of Convector No. 36, which had a heating unit only 2 in. in height, well marked increments of heat output occurred when the widths of the inlets were increased above 5 in. In the case of the grilled inlets it was probably necessary to increase the free area up to a value comparable to the area of a 5-in. ungrilled inlet before the resistance of the heating unit became the controlling factor. Evidently this value was not reached with any of the limiting widths of grilled inlets attainable, because the increments of heat input showed no marked decrease as the widths of these inlets were increased above 5 in.

The results shown in Fig. 6 indicated that the relations between the free areas of the inlets and the free areas of the heating units, or more restricted portions of the convector, might be of more significance than the widths of the inlets. The heat outputs were therefore plotted against the free areas of the inlets, and the results are shown in Fig. 7. These curves have the same general shape as those shown in Fig. 6, but the relative positions of the curves for the grilled and ungrilled inlets are reversed, owing to the fact that for given widths the free areas of the grilled inlets were considerably less than the areas of the ungrilled inlets. In the case of each convector the heat output obtained with a given free area was greater with a grilled inlet than with an ungrilled inlet. The difference was small, never amounting to more than 3 per cent, and may have resulted from somewhat greater turbulence caused by the presence of the grille. Such increased turbulence would have a tendency to increase the heat transfer. The somewhat better performance with the grilled inlets may also have resulted from the fact that in the case of an ungrilled inlet a considerable portion of the air could flow around the upper edge of the opening and pass up through the front part of the heating unit, thus making this part of the unit more effective than the back, while with the grilled inlet the air entered over the whole face in jets and more of it was forced toward the back of the cabinet, thus increasing the effectiveness of the back part of the heating unit.

The curves shown in Fig. 7 tend to confirm the hypotheses made in connection with the discussion of Fig. 6. In the cases of Convectors Nos. 35 and 37, increasing the width of the ungrilled inlet beyond 4 or 5 in. resulted in no material increase in the heat output, indicating that with the greater widths of inlets the resistance of the heating unit became the dominant factor in influencing the heat output. The free areas through the heating units and the free areas of the outlets are shown in Fig. 7. The free area through the heating unit was regarded as the total unobstructed area of the inside of the cabinet at a horizontal section through the most obstructed plane in the heating unit. Each cabinet fitted the heating unit quite closely, thus preventing any bypassing of air around the sides and ends of the unit. It may be observed that the curve for Convector No. 35 with the ungrilled inlets tended to flatten after a free area of approximately 129 sq in. was reached, corresponding to a width of inlet of 4 in. For Convector No. 37 this flattening occurred at a free area of approximately 135 sq in., corresponding to a width of inlet of 5 in. Comparing these values with the free areas through the heating units, it is evident that for Convector No. 35 the critical width was accompanied by a ratio of free area of the inlet to free area of the heating unit of 1.39. The corresponding ratio for Convector No. 37 was 1.25. For Convector No. 36 no marked change in the slope of the curve appeared even with a ratio of 2.1, indicating that the critical width of ungrilled inlet was greater than 6¾ in. The critical ratio is apparently related to

the height of the heating unit, since with Convectors Nos. 35 and 37, having 6-in. and 5¼-in. heating units respectively, ratios of the same order of magnitude were obtained, while with Convector No. 36, having only a 2-in. heating unit, the ratio was considerably greater. The correlation between the critical ratios and the heights of heating units was not exact, because undoubtedly factors other than height and free area, such as arrangement of fins, affect the frictional resistance of the heating unit.

It may be further observed from Fig. 7, that, in the case of grilled inlets, the curves also show some tendency to change slope with free areas of inlets of approximately 95 and 110 sq in. for Convectors Nos. 35 and 37 respectively. These both correspond to a ratio of free area of the grilled inlet to the free area of the heating unit of 1.02. In the case of Convector No. 36, with only a 2-in. heating unit, the ratio was again greater, having a value of 1.17. The results shown in Fig. 7 therefore indicate that in the case of convectors having heating units approximately 6 in. in height, as in Convectors Nos. 35 and 37, the free area of an ungrilled inlet should be approximately 25 per cent in excess of the free area through the heating unit, and no appreciable gain in heat output can be obtained by increasing the free area of the inlet materially above 25 per cent in excess of the free area of the heating unit. In the case of convectors having heating units of approximately 2 in. in height, the effective free area of an ungrilled inlet is limited only by the distance between the floor and the bottom of the heating unit. The free area of grilled inlets for all convectors should be at least equal to the free area of the heating unit.

Effect of Width of Outlet

The heat outputs obtained from Series III and Series IV, using a constant width of ungrilled or grilled inlet with different widths of grilled outlets, are shown in Fig. 8, which was developed by the same method as that employed for Fig. 6. The curves in Fig. 8 all present the same characteristics, in that as the width of outlet was increased the heat output also increased, but reached a distinct maximum when the width of the outlet was 5 or 6 in. Increases above 5 or 6 in. in width were accompanied by more or less marked decreases in heat output. With Convectors Nos. 35 and 37 the substitution of the grilled inlet had no material effect on the heat output for widths of outlets up to 5 in., while with Convector 36 such substitution had a marked effect over the whole range.

Several explanations may be offered for the tendency of the heat output to reach a maximum with successive increases in the width of the outlet. The width was increased by lowering the bottom edge of the outlet opening, and with increasing widths of outlets this procedure may have reduced the effective stack height, and thus somewhat reduced the flow of air through the heating unit. It is rather difficult, however, to determine just what does constitute effective stack height in any given case.

In any stack with a front outlet similar to a convector cabinet the air tends to crowd to the top before making the turn out of the outlet. This condition results in high air velocities toward the top of the outlet and the upper portion becomes more effective than the lower in passing air. As the bottom of the outlet is lowered, the high velocity portion of the air stream tends to remain approximately the same width, and the lower portion of the opening becomes less and less effective. Finally a width of outlet is reached which is just equal to the width of the air stream, and if the width is now increased by lowering the bottom edge, air from the room will enter the lower part of the opening. This air will be drawn into the main air stream and be discharged through the upper portion of the opening without having passed through the heating unit. This interference tends to reduce the flow of air and thus to reduce the heat output of the convector. Hence, beyond a certain width, the increases in width would merely permit more air to be drawn in at the bottom of the opening, and progressive increases in width would be accompanied by decreases in the heat output.

The heat outputs obtained from Series III and IV were plotted against the free areas of the outlets, and the results are shown in Fig. 9. Since only grilled out-

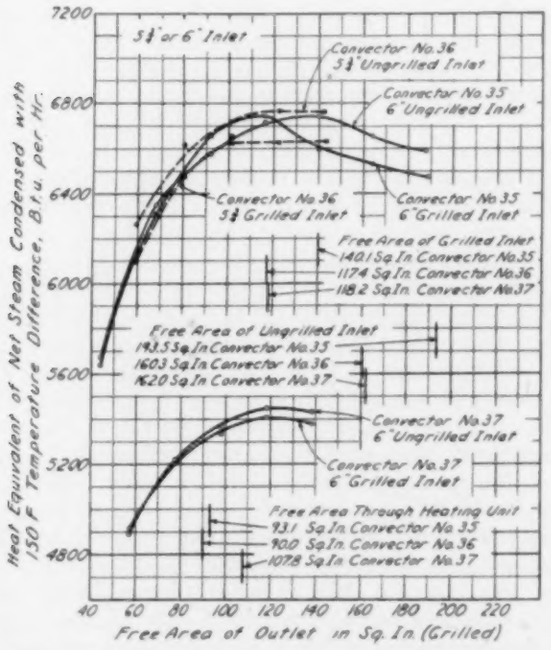

5¼" or 6" Inlet
Convector No. 36
5¼" Ungrilled Inlet
Convector No. 35
6" Ungrilled Inlet
Convector No. 36
5¼ Grilled Inlet
Convector No. 35
6" Grilled Inlet
Free Area of Grilled Inlet
140.1 Sq. In. Convector No. 35
117.4 Sq. In. Convector No. 36
118.2 Sq. In. Convector No. 37
Free Area of Ungrilled Inlet
193.5 Sq. In. Convector No. 35
160.3 Sq. In. Convector No. 36
162.0 Sq. In. Convector No. 37
Convector No. 37
6" Ungrilled Inlet
Convector No. 37
6" Grilled Inlet
Free Area Through Heating Unit
93.1 Sq. In. Convector No. 35
90.0 Sq. In. Convector No. 36
107.8 Sq. In. Convector No. 37
Heat Equivalent of Net Steam Condensed with 150 F Temperature Difference, B.t.u. per Hr.
Free Area of Outlet in Sq. In. (Grilled)

Fig. 9—Heat output of convectors with outlets having different free areas

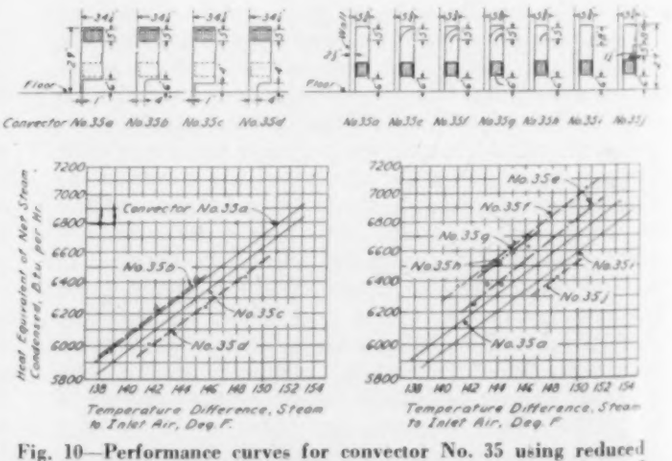


Fig. 10—Performance curves for convector No. 35 using reduced length of inlet and different arrangements of vanes at inlet and outlet

lets were used, there was a fixed relationship between the free area and the width of the grille. Hence, the curves in Fig. 9 present the same characteristics as those shown in Fig. 8. In addition, however, the free areas of the inlets and of the heating units are shown in Fig. 9. It is probable that the heat output of a convector is influenced by the relative resistances of the outlet and the heating unit, just as it was shown to be by the relative resistances of the inlet and the heating unit. Some indication of this condition is shown by the curve representing the results obtained from Convector No. 35 when used with a grilled inlet. This convector had a 6-in. heating unit, and when used with a grilled inlet it probably offered the greatest combined resistance of any of the convectors. In this case the reduction in heat output with successive increments in the width of the outlet in excess of 5 in. or free area in excess of 116 sq in. occurred at a greater rate than it did in the case of any other arrangement. Further indication is given by the curves for Convector No. 36. This convector had a heating unit only 2 in. in height, and offered the least resistance of any of the convectors. In this case the substitution of a grilled inlet for an ungrilled inlet resulted in a marked reduction in heat output over the whole range, indicating that the resistance of the heating unit was not sufficient to become the controlling factor and thus allowed the additional resistance offered by the grille to be directly reflected in a reduction in heat output. In Convectors Nos. 35 and 37 the resistances of the heating units were sufficiently high to prevent the small additional resistance of the grille at the inlet from being directly reflected in reductions in the heat outputs.

A study of the ratios of the free areas of the outlets to the free areas of the heating units at maximum heat output as shown in Fig. 9 failed to prove any very definite correlation between the areas themselves or between the areas and the heights of the heating units; further than that the free area of the outlet should not be less than 10 per cent and probably not greater than 30 per cent in excess of the free area of the heating unit. The one exception proved to be Convector No. 35 when used with an ungrilled inlet, in which case the ratio for maximum heat output was 1.45. The ratios of the free areas of the inlets used with the three convectors to the free areas of the heating units varied from 1.1 to 1.5 for the grilled inlets, and from 1.5 to 2.1 for the ungrilled inlets. Hence, the resistances of the inlets used with the different widths of outlets did not have any material influence on the maximum heat output, and the ratios of the free areas of the outlets to the free areas of the inlets do not seem to have any particular significance in this study.

Effect of Reduced Length of Inlet

In order to conceal piping connections it is more or less common practice for the length of the inlet to be from 6 to 8 in. shorter than the overall length of the convector cabinet. Studies were made to determine whether this practice entailed a sacrifice in heat output, and the results, plotted to logarithmic scales, are shown in the left half of Fig. 10. Details of the different arrangements are shown in the insets at the top of the figure. Curve No. 35*a* shows the heat outputs obtained from Convector No. 35 with an inlet 32¼ in. in length and 6 in. in width, while Curve No. 35*b* shows the heat outputs obtained when the inlet was shortened 6 in. by reducing it 3 in. at each end. Curves Nos. 35*c* and 35*d* respectively show similar results obtained with an inlet 4 in. in width. It may be observed that with the inlet 6 in. in width no sacrifice in the heat output resulted from the reduction in length. The small apparent gain indicated probably represents the limit of the ability to exactly duplicate tests rather than an actual increase in heat output. In the case of the 4-in. inlet, however, the reduction in the length of the inlet was accompanied by a reduction of approximately 1.5 per cent in heat output at the standard temperature difference of 150 F. It is probable that the same criterion should be applied to the reduction in length as that developed for the reduction in width of inlet, namely that the free area of an ungrilled inlet should be at least 25 per cent in excess of the free area through the heating unit.

Effect of Directing Vanes

The curves in the right half of Fig. 10 show the results obtained when vanes of different types were installed at the inlet or outlet of a convector cabinet. Convector No. 35, having a 6-in. ungrilled inlet and a 5-in. grilled outlet, was used as a base construction and the details of the modifying vanes are shown in the various insets. The basic cabinet is designated as Convector No. 35*a* and the corresponding curve of heat outputs as Curve No. 35*a*. The addition of the vane in the top of the cabinet, Convector No. 35*e*, resulted in a slight improvement in the heat output, as shown by Curve No. 35*e*. This improvement amounted to approximately 1½ per cent at the standard temperature difference of 150 F. The use of a vane at the center in addition to one at the top, Convector No. 35*f* and corresponding result Curve No. 35*f*, resulted in a gain of approximately 4 per cent at the standard temperature difference of 150 F, as compared with the heat output of Convector No. 35*a* used without vanes. The removal of the vane at the top, thus converting Convector No. 35*f* into Convector No. 35*h*, did not result in an appreciable decrease in heat output as compared with that obtained with the two vanes in Convector No. 35*f*. The addition of a vane at the inlet, converting Convector No. 35*f* into Convector No. 35*g*, resulted in a slight decrease in heat output, indicating that nothing is to be gained by using vanes at the inlet of a convector.

The conditions shown by Curves Nos. 35*a* to 35*h* can be readily explained by the character of the air flow. In the case of a front outlet similar to the one in Convector No. 35*a*, the air tends to crowd to the top and the lower portion of the outlet is only partially effective in passing air. The banking of the air in the upper corner forms a more or less dead air space that tends to serve as a directional surface without the necessity for a vane. Using a vane in this location may give the air flow slightly better direction, but does not correct the ineffectiveness of the lower part of the face of the outlet. Hence no great improvement is effected by the addition of this vane, as shown by the comparison between Curves Nos. 35*a* and 35*e*, and Curves Nos. 35*h* and 35*f*. When the vane is placed at the center, however, the air flow is directed more uniformly over the whole face of the outlet, and the lower part becomes

effective. Hence, this arrangement offers the best possibility for material improvement over the outlet with no vanes, as substantiated by Curves Nos. 35*h* and 35*a*. Again it may be observed by comparing Curves Nos. 35*h* and 35*f* that the addition of the vane at the top of the convector cabinet resulted in no material gain in heat output. The air flow into an inlet is more or less uniform over the whole face. Hence, the addition of a vane at the inlet is more liable to act as a detriment by introducing additional friction than it is to serve as a guide to give more uniform air flow over the heating unit.

The resistance to air flow around a bend or out of a front or side outlet is to a much greater extent determined by turbulence existing at the inside edge or corner, than it is by conditions existing at the outside corner. Conditions at the inside corner can sometimes be improved by streamlining or creating a venturi throat on the down-stream side of the turn. An attempt was made to improve the heat output obtained with the 8-in. width of outlet used with Convector No. 35 by the use of such a venturi throat. The details of this throat are shown in the inset in the right half of Fig. 10. It should be noted that it was not possible to place this vane on the down-stream side, since this would have located it outside of the outlet. Hence, it was placed on the up-stream side. In the case of elbows this construction has been shown to have some merit, but it is not nearly as effective as a venturi on the down-stream side. The basic curve for Convector No. 35 with an 8-in. outlet is shown as Curve No. 35*i* in Fig. 10. The use of the venturi throat resulted in a net loss in heat output, as shown by Curve No. 35*j*. Hence, the use of such an arrangement is not to be recommended.

Effect of Different Locations of Inlets and Outlets

The cabinet furnished with Convector No. 31*a* had two inlets, one in the front and one in the back of the cabinet. With the back inlet blocked this convector was designated as Convector No. 31. Details of these arrangements are shown in the insets in the lower part of Fig. 11. The performance curves plotted to logarithmic scales for Convectors Nos. 31 and 31*a* in Fig. 11 show that when the back inlet was blocked the heat output at the standard temperature difference of 150 F was reduced approximately 4 per cent. Hence, in this case the use of the double inlet could be justified. It should be noted, however, that in this particular convector the front inlet was only 3½ in. in width, and the extra area could probably just as well have been obtained by making this inlet wider instead of using a second inlet. A double inlet, however, might be advantageous in the case of a convector with the heating unit located close to the floor.

Studies were made on two convectors to determine whether any improvement in heat output could be effected by using a top outlet instead of a front outlet having the same free area. Convector No. 33 furnished with a commercial cabinet and Convector No. 35*c* equipped with a special cabinet were used for this purpose, and with the modified cabinets having top outlets were designated as Convectors Nos. 33*a* and 35*k* respectively. The details of these arrangements are shown in the insets in the upper part of Fig. 11. The results shown in Fig. 11 indicate that a marked improvement in heat output resulted from the use of a cabinet having a top outlet instead of a front outlet. In the case of Convector No. 33 the gain in heat output effected by the use of the top outlet was 14.5 per cent at the standard temperature difference of 150 F, while in the case of Convector No. 35*c* the gain was 9.5 per cent. Undoubtedly the amount of gain depends on the relative resistances of the heating unit and the original front outlet, and on the relative active stack heights involved. Sufficient data were not available, however, to permit of any accurate evaluation of the effects of these additional factors.

Effect of a Humidifying Pan on the Heat Output

The construction of a convector suggested that it might be modified to serve as a device for both heating and humidifying the air. Convector No. 11 was therefore adapted to humidify the air by installing a water pan having a false bottom. The space between bottoms was connected to the steam space in the heating unit by tubes at the ends, as shown in the inset in Fig. 12. The original 4¾-in. outlet was increased to a width of 8 in. and the modified convector was designated as Convector No. 11*a*.

The performance curve for the original convector, No. 11, is shown as the lowest curve in Fig. 12. The total heat outputs, corresponding to the total steam condensed by both the heating unit and the water pan, are shown for Convector No. 11*a* by the top curve in Fig. 12. Apparently this total heat output was more or less independent of the temperature difference between the steam and the air entering the inlet of the convector. The total amount of water evaporated remained practically constant at a value of 2.0 lb per hour, or an evaporation of 2.6 lb per square foot of surface per hour. The water temperature varied from approximately 190 F at a temperature difference of 137 F between steam and enter-

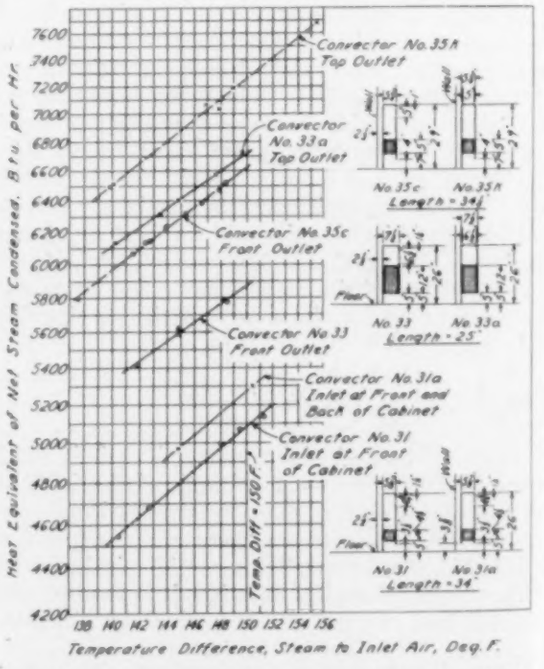

Heat Equivalent of Net Steam Condensed, Btu per Hr
Temperature Difference, Steam to Inlet Air, Deg. F.
Temp. Diff. = 150 F
Convector No. 35A Top Outlet
Convector No. 33a Top Outlet
Convector No. 35c Front Outlet
Convector No. 33 Front Outlet
Convector No. 31a Inlet at Front and Back of Cabinet
Convector No. 31 Inlet at Front of Cabinet
No. 35c
No. 35A
Length = 34"
No. 33
No. 33a
Length = 25"
No. 31
No. 31a
Length = 34"
Floor

Fig. 11—Performance curves for convectors Nos. 31, 33 and 35 with different arrangements of inlets and outlets

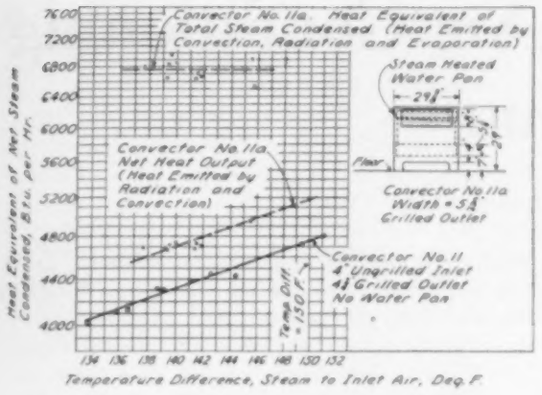

Convector No. 11a. Heat Equivalent of Total Steam Condensed (Heat Emitted by Convection, Radiation and Evaporation)
Steam Heated Water Pan
Convector No. 11a Net Heat Output (Heat Emitted by Radiation and Convection)
Convector No. 11a Width = 5⅛ Grilled Outlet
Convector No. 11 4" Ungrilled Inlet 4½ Grilled Outlet No Water Pan
Temp. Diff. = 150° F.
Heat Equivalent of Net Steam Condensed, B.t.u. per Hr.
7600
7200
6800
6400
6000
5600
5200
4800
4400
4000
134 136 138 140 142 144 146 148 150 152
Temperature Difference, Steam to Inlet Air, Deg. F.

Fig. 12—Effect of installing steam heated water pan at outlet of convector No. 11

ing air to approximately 180 F at a temperature difference of 150 F. Based on one air change per hour, with a temperature of 70 F indoors and zero outdoors, the continuous evaporation of 2.0 lb per hour is sufficient to maintain approximately 5800 cu ft of space at a relative humidity of 35 per cent. Hence, Convector No. 11*a* may be regarded as a very effective humidifier. The air leaving the convector was not saturated and no evidence of moisture condensing on the cabinet was observed.

The effectiveness of Convector No. 11*a* as a heating device, including heat added to the air and heat radiated to the room, was obtained by subtracting the latent heat of evaporation at the temperature of the water in the pan from the heat equivalent of the total steam condensed. The heat outputs thus obtained are shown by the intermediate curve in Fig. 12. It may be observed that, at the standard temperature difference of 150 F, the effective heat output of Convector No. 11*a*, with the water pan, represented an increase of approximately 8 per cent over the heat output of the original Convector No. 11, without the water pan. This increase in heat output can probably be explained by the fact that the superficial area of the pan containing water at from 180 F to 190F was at a higher temperature than that of the air passing over it. It thus became effective heating surface in addition to the normal heating surface of the heating unit, the bottom of the pan and the connecting pipes which were at steam temperature.

Conclusions

The following conclusions may be drawn as applying to the data obtained from these tests:

(1) From the standpoint of heat output the free area of a grilled inlet should be at least equal to, and of an ungrilled inlet should be approximately 25 per cent in excess of, the free area through the heating unit.

(2) For convectors having heating units 5 or 6 in. in height, or those having heating units with comparatively high frictional resistance, no material gain in heat output can be obtained by increasing the free area of an ungrilled inlet to more than 25 per cent in excess of the free area through the heating unit.

(3) For maximum heat output the free area of the outlet of a convector should not be less than 10 per cent and probably not greater than 30 per cent in excess of the free area through the heating unit.

(4) The most effective position for a directing vane at the outlet of a convector cabinet is in a central location with respect to both the cabinet and the outlet face.

(5) A directing vane located at the top of a convector cabinet is effective to a very small extent.

(6) Directing vanes at the inlet to a convector cabinet are not effective and may prove to be detrimental by introducing additional friction.

(7) The heat output of a convector having a cabinet with a top outlet may be from 9.5 to 14.5 per cent greater than the heat output from the same convector with a cabinet having a front outlet with the same free area as that of the top outlet.

(8) A convector may be converted into an effective humidifier without loss in effectiveness as a heating device by installing in the top of the cabinet a water pan provided with a steam space connected to steam headers in the heating unit.

Acknowledgments

The data presented in this paper were obtained in connection with an investigation conducted by the Engineering Experiment Station of the University of Illinois, of which M. L. Enger, dean of the College of Engineering, is the director. The investigation was conducted in the Department of Mechanical Engineering, of which O. A. Leutwiler, professor of Mechanical Engineering Design, is the head. The results will ultimately comprise part of a bulletin of the Engineering Experiment Station.

Engineering Tests of Mineral Wool

In an address delivered on Mineral Wool Day at the New York World's Fair July 13, 1939, Prof. F. B. Rowley, director of the Experimental Engineering Laboratory of the University of Minnesota, stated that one of the latest properties in mineral wool to receive scientific investigation is that of moisture resistance. Simple tests have demonstrated that mineral wool is not hygroscopic and, therefore, will not absorb moisture from the air; however, there are other factors which have entered into the problem in recent years that make the problem more complicated than can be settled by a simple test. Recent studies conducted at the University of Minnesota, he said, have summarized the moisture condensation problem into a few factors, most important of which are, temperature and humidity of air within a building, outside air temperature, resistance to vapor of various parts of the wall, and the thermal properties of the materials within the wall. In most types of building construction vapor will pass through the wall in direct proportion to the vapor pressure difference across the wall, and in inverse proportion to the vapor resistance along the path of vapor flow. For new construction the best solution of the problem is to prevent the vapor from entering the wall by using some vapor resisting material on the inner section of the wall.

Another project which has been attacked in a practical way involves the relation of insulation to comfortable summer living conditions. Duplicate test houses have been constructed in Minneapolis and New York, and each is equipped with recording instruments for determining the effects of sunshine, air temperature, wind velocity, and wind direction on the comfort conditions maintained within the structures.

Fire Protection for Air Conditioning Systems

By R. C. Loughead,* Detroit, Mich.

IT is presumed that members of the society have all the knowledge and facilities necessary to install an air conditioning system so that it will function properly and efficiently and provide clean, fresh air at the proper temperature and humidity; and that the fire prevention engineer knows little about the engineering problems of the air conditioning profession. It is known, however, that with proper design and planning, the potential fire hazard associated with air conditioning may be minimized and to a large extent eliminated and fires entailing loss of life may be prevented.

The regulations of the *National Board of Fire Underwriters,* (pamphlet No. 90 with revisions of May, 1939) cover the installation of air conditioning, warm air heating, air cooling and ventilating systems, and these regulations provide for the mitigation of the fire hazards in connection with air conditioning. It is the purpose of this paper to endeavor to point out the necessity of regulation based on fires which have already occurred due to faulty design and installation and to provide heating, ventilating and air conditioning engineers with sufficient information relative to the inherent fire hazards which exist with respect to air conditioning so that systems may be more generally properly designed and installed from a fire prevention standpoint.

A system installed in a large department store provides conditioned air to the sub-basement, basement, first floor, and mezzanine, with the conditioning equipment located in the sub-basement. Such an installation, regardless of its design, might include more potential fire hazards than any other type since it serves more than one floor from below so that gravity, in conjunction with the direction of air flow, tends to increase the possibility of combustible materials entering the recirculating ducts. But this particular system is installed according to the regulations of the *National Board of Fire Underwriters* and is therefore presumed to be properly protected. The ducts are constructed of incombustible material, they are accessible for inspection and cleaning, protected where in the vicinity of combustibles, lined and lagged in part with incombustible materials, and provided with automatic fire dampers at floors, fire walls, and partitions.

The recirculating duct openings are covered by screen of small mesh so that it is impossible for other than dust or lint to enter these ducts. The filters are incombustible, kept clean, and protected by a sprinkler system actuated by a manually reset thermostat which is also connected to automatic dampers, cutting off the plenum chamber and stopping the fan. Steam coils for heating are properly protected, and brine is utilized for cooling. This is a standard system, installed according to the regulations of the *National Board of Fire Underwriters,* and there have been no fires in connection with an air conditioning system installed according to these standards.

Generally speaking, the system described is a paradox, a "pipe dream," for such a system with the protection described, is practically non-existent, although these essential installation features are necessary if life and property are to be protected from fire in air conditioned buildings. That such a protected system does not exist to a greater degree is not the fault of engineers who design and install but rather the responsibility of the fire prevention engineer who has failed to educate them to design with proper fire prevention and protection features. This safe design will more generally follow when the lessons learned from fires which have occurred are thoroughly understood, and this can be accomplished at a relatively small increase in cost. But before discussing fires which will serve as illustrations of improper design or faulty maintenance, it might be well to point out the different parts of an air conditioning system in which potential fire hazards may exist.

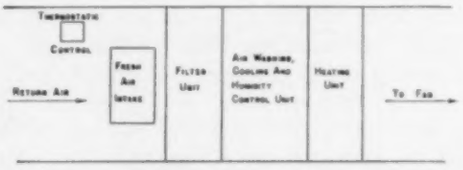


Fig. 1—Relative location of appurtenances

A typical air conditioning system may consist of the following parts: a fan producing the movement of air; a system of ducts located on the discharge side of the fan, used for conveying the conditioned air to the desired locations; a return system of ducts from these locations back to the conditioning units; a fresh air intake for supplying additional air from the outside; a series of filters in the return and fresh air ducts for cleaning the air; a water spray for washing, humidifying, and cooling the air; steam coils for raising the temperature; refrigeration equipment for cooling the water spray or for cooling the air directly or indirectly by means of coils in the ducts; and the valves, controls, and other appurtenances for the automatic operation of the system. Fig. 1 shows a schematic diagram of the relative arrangement of the component parts.

The fan or blower should be so located and arranged that ready access is afforded for repairing, cleaning, lubrication, and inspection. One or more controls, readily accessible, should be provided so that the fan may be promptly shut down in case of fire and an automatic fire detecting device should also perform this function. Due to the relatively high air velocity in the ducts,

*Chief Engr., Michigan Inspection Bureau.
Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Mackinac Island, Mich., July, 1939.

sometimes as high as 1200 fpm, materials which become ignited burn more vigorously on account of the excess oxygen.

The discharge ducts which convey the treated air to the areas conditioned may consist of many feet of ducts of various sizes, terminating in meshed or grilled openings, usually located at a considerable distance above the floor, which, in combination with the direction of air flow, generally precludes the possibility of bits of paper or other combustibles entering these openings. However, more or less dust may collect therein, depending upon the efficiency of the filters. Certain types of filters which permit the passage through them of oil soaked lint should not be used since there is a possibility of combustible material lodging in these ducts, and in any event means for periodic inspection and cleaning should be provided. In general, the inherent hazard of the discharge ducts is relatively small, but they may be the means of carrying fire, hot gases, and refrigerants into the conditioned areas from other points and therefore may require protection.

The return or recirculating duct system is usually as extensive as the discharge ducts and, if improperly designed and installed, may present a fire hazard much greater than any other part of the system. When the openings from the conditioned areas are at or near the floor, this, together with the direction of the movement of air, may invite the tossing of burning cigarettes, bits of paper, and other combustible debris through the grille openings; therefore, the return ducts should be provided with inspection and clean-out openings so that they may be kept clean. The mesh of the screen or grilles should be made as small as possible without reducing the efficiency of the system, and in new designs this may be accomplished by increasing the duct inlet areas. Recognition should always be given to the added possibility of a fire originating anywhere in the conditioned area being drawn through these ducts, involving any combustibles within the system and then being conveyed into all parts of the building.

Each conditioning system is usually provided with a fresh air inlet, which should be of fire resistive construction, used for no other purpose, and the opening to the outside should be located well away from any possible exposure or alleys and areaways where burning materials or smoke may be taken into this duct and distributed throughout the system.

In general, when a duct system is installed at the time the building is erected, the ducts may be concealed in the structure, but otherwise the vertical ducts are usually installed in the building columns and the horizontal ducts suspended from the ceiling. In any event, all ducts should be constructed of incombustible materials and well separated from woodwork and other combustibles. When lining or lagging is used for the minimizing of heat losses or the reduction of noise, this lining or lagging should be of incombustible material to eliminate the fire hazard due to combustible linings. Even though the materials used may not be highly combustible under ordinary conditions, the possibility should be recognized that they may burn vigorously in the presence of the air movement in the ducts.

There are a number of different types of filters, only a few of which are entirely incombustible, but in any event their function is to collect dust and lint which are combustible. Some filters are made up of units of steel wool, spun glass, rock wool, or a series of screens or metal surfaces set into frames and in order that they may be efficient they may be dipped or sprayed with a high flash oil whose function it is to trap the dust particles. Another type consists of a revolving screen which passes through a reservoir of oil. The air stream type consists of a series of staggered rods covered with oil, the oil picking up the dust. This type may present an added hazard, allowing an unusual amount of oil soaked dust and lint to pass through the filter and be deposited in the discharge ducts. All of these types are more or less hazardous, even when clean, on account of the oil. Another type is built in units consisting of cloth and cotton or cellulose so arranged in a zig-zag manner as to present a large surface to the air, and while no oil is required in connection with these filters, they are combustible and a source of fuel should a fire reach them. A cloth of spun glass, which is incombustible, is now available for this type of filter and is probably the least hazardous of any filter on the market at the present time. The ultimate aim of the filter manufacturer is to discover a liquid which will not burn and at the same time possess all the filtering qualities of oil.

Steam coils located in the main duct for heating the air are usually of the fin type and should be located well away from combustible materials.

Refrigeration in connection with air conditioning may be used for cooling water sprays, for cooling brine which is circulated in coils in the ducts, or the refrigerant may be expanded directly through these coils, in which case the failure of the coils releases the gas into the duct system and in turn to the conditioned areas. Many of these refrigerant coils are made up with solder which generally has a relatively high melting point but nevertheless would fail under actual fire conditions, releasing the gas which may be explosive or toxic.

The purpose of the description given is to show that there may be a decided fire prevention problem in connection with air conditioning and to recommend the regulations of the *National Board of Fire Underwriters* for the protection of these hazards. In addition to these safeguards, constant vigilance should be practiced so that the chance of a fire involving the entire system is remote. Cleanliness is of paramount importance.

For some time the Underwriters' Laboratories have listed refrigeration units for use in connection with air conditioning, but it is only relatively recently that they have developed a fire test procedure for filters and now they are listing unit filters under the following classifications:

Class I filters are those which when clean do not contribute fuel when attacked by flames and which emit only negligible quantities of smoke.

Class II filters are those which when clean burn moderately when attacked by flames or emit moderate amounts of smoke, or both. These filters may be consumed to some extent but will not project flame or sparks down the duct during the test.

This means that filters which are readily flammable or which when tested project flame or sparks downstream, or which emit more than moderate quantities of smoke or emit products of combustion which are unusually disagreeable or toxic, are not considered as suitable for

listing by the Underwriters. Quoting from the Laboratories standard, the test equipment is as follows:

The duct used in the test is of galvanized iron construction, reinforced with angle irons. It is 21 in. wide by 41 in. deep and 27 ft 10 in. in horizontal length. At one end it is tapered to connect to the discharge of a variable speed, electrically driven blower. The other end of the duct is an open discharge. At about the middle of the duct are frames to receive two 20 by 20 in. filter units. A tight fitting door permits access to the filter frames and numerous glazed openings permit observation of both sides of the filters and of conditions in the duct downstream from the filters. Four 1-in. elbows form burner outlets for the city gas fuel, so arranged that two flames play on each filter. The flames are yellow and wavering and extend intermittently about 12 in. beyond the filter frames when the latter are empty. The burners consume about 9 cu ft of gas per minute, having a Btu value of about 805 per cu ft.

Research Needed

Several manufacturers are developing equipment for the detection of smoke, and the *National Board of Fire Underwriters* is sponsoring tests and research to determine effective means of detecting and controlling spread of smoke in air conditioning systems and to reduce the possibilities of smoke damage and panic hazard. These devices are intended to be used to shut down blowers, close dampers, sound alarm, and perform other necessary functions to prevent the spread of fire and smoke by the air conditioning system. Such a system has been installed in three large air conditioning systems in a Detroit department store and will be supervised and regularly inspected and maintained by a central station company operating in that city.

One of the basic principles of fire prevention engineering is the division of a building into fire areas by fire walls and protection for all communicating or vertical openings so that the loss in any one fire may be minimized. It has long been realized that air conditioning which entails the piercing of these fire walls and floors with ducts would offset much of the effort which has been made to reduce the areas which may be involved in a fire. The actual regulation of theaters, motion picture studios, paint spraying, etc., came about only after there was loss of life by fire in these occupancies, and it is hoped that lessons such as these will not be necessary in order that the installation and use of air conditioning may be properly regulated. While there is no record of loss of life associated with improperly designed systems at this time, the fire loss and smoke damage has been considerable.

Carelessness Causes Fire

A fire occurring in a Detroit department store over a year ago, causing a loss of about $22,000 illustrates the fire hazard in connection with improper type filters. In this case, the filter involved consisted of a revolving screen passing through a reservoir of oil, which was carelessly ignited by an acetylene torch. After the fans were stopped, most of the smoke and heat, very fortunately, backed up through the fresh air intake shaft and away from the interior of the store, reducing the loss to an appreciable extent. This does not mean that filters of this type should be condemned, but that, on account of the large quantities of oil, they are not adapted to occupancies where the panic hazard is pronounced. Other fires show that combustible filters, incombustible filters covered with oil, or even filters entirely incombustible but loaded with dust and lint which may become ignited, may be a source of danger, and should be suitably protected. Records of still other fires illustrate the fact that while filters may be a potential fire hazard, they do not necessarily constitute the greatest fire hazard, which may be associated with other substandard features.

Two fires occurring in widely separated localities, New York and California, were caused by combustible duct lining. In the one in New York City the burning of rubbish in an areaway directly under the discharge duct of the air conditioning system of a theater caused smoke, sparks, and heat to be discharged into the balcony. Fortunately there was only a small audience when the fire started. The other fire occurred in a Los Angeles office building, and when the janitor discovered that the air conditioning system was emitting smoke due to burning duct lining, it was necessary for him to go from the basement to the roof of this 10-story office building in order to shut down the fan. Proper design and reasonable foresight would have made this unnecessary.

In March of this year a $150,000 loss occurred to a school building in Minnesota which might have been minimized or entirely eliminated with proper design of both structure and air conditioning system. This school building was three stories high, of fire resistive construction except the roof, all floors and the ceiling over the third floor being of reinforced concrete construction. The conditioning equipment was installed in the roof space between this incombustible ceiling and the combustible roof. The discharge ducts to all the rooms were connected to a main discharge duct in this roof space, but the recirculating or foul air ducts exhausted directly into the roof space, not being directly connected to the heating system. All ducts throughout the building were of incombustible construction except the horizontal ducts in the roof space, which were combustible. The fire, which originated in the roof space, not only was transmitted to the school rooms through the discharge ducts, but burning brands from the roof boards dropped through the foul air ducts into the classrooms. It is entirely possible that proper design would have prevented this fire.

Last winter in a city suburban to Detroit, unsold Christmas trees were being burned in an alley and smoke from the fire, entering the fresh air duct of a theater air conditioning system, caused a small smoke loss which might have been greater had the fan been operating or had the source of the smoke not been promptly discovered.

Fire starting in a pile of kindling wood outside a school building in Texas a number of years ago entered an air intake to the ventilating system and completely destroyed the building. At a textile plant in Holyoke, Mass., smoke and soot from a burning tar kettle in the yard were drawn through the fresh air intake of an air conditioning system into several manufacturing rooms on the third, fourth, and fifth stories of the building. Soot deposited on worsted yarn in process caused stoppage of plant operations and required several hours' time to clean off equipment and restore the plant to operating condition. Whether these fires were due to the improper location of the fresh air intakes or the improper burning of rubbish or placing of a tar kettle is a controversial

subject; the fact remains that had the conditioning systems in these particular buildings been properly designed, installed and maintained, there would have been no fire or smoke loss.

In a village suburban to Detroit, considerable smoke loss occurred in a dwelling due to a forced air heating system. This new home was not as yet occupied and a sanding machine had been placed on the return duct inlet located in the floor of the living room. A fire in the sander resulted in smoke being conducted by means of the discharge ducts into every room in the building in sufficient quantities so that it was necessary to completely redecorate although the flames themselves did little damage.

Undoubtedly the fire which has caused the most comment was the one occurring in January of last year in the air conditioning system of a large Detroit department store. The system was located in the sub-basement and conditioned this area, the basement, the first floor, and the mezzanine. On the evening of the fire a number of employees were taking inventory and at the time of the fire, which occurred at 10:15 p. m., the blower fans had been shut down for over an hour. Due to an unknown cause, a fire originating in the duct system caused large quantities of smoke and some flame to be discharged into the basement and sub-basement through both outlet and inlet ducts. A total of fifteen sprinklers opened, controlling the fire. The damage, which was almost entirely due to smoke, amounted to about $110,000, and the fire department is reported to have taken several employees down ladders from the second story windows. This fire illustrates very graphically the necessity for improved installations to eliminate the possibility of a pronounced panic hazard as well as heavy contents loss in connection with this type of occupancy. It is not difficult to picture a catastrophe if a sale had been in progress during the normal operation of the system.

This fire was evidently fed by an accumulation of waste paper scraps, lint, and other refuse in both the discharge and return ducts and might have been ignited by a cigarette being carelessly thrown into a return duct or the plenum chamber. In any event, this combustible material could not have accumulated in the duct system with properly designed intake grilles and proper air filters. Only a source of ignition was necessary to start a fire and could have been prevented had the duct system been kept clean. An examination of parts of the horizontal ducts in the discharge system and not involved in the fire showed an accumulation of oil soaked lint and dust covering as much as one-third of the duct floor to the extent of ¼ to ½ in. deep. This lint and dust had passed through the filter, and sales slips, gum wrappers, cigarette butts, and other combustible materials were found in the horizontal return ducts.

It can be seen from these descriptions of fires and a study of the present regulations that the fire hazard in connection with air conditioning and ventilating systems may be great or small, depending upon the safeguards provided and the inherent hazards due to design. The system supplying two or more floors from below should receive special attention, while the system supplying one floor or part of a floor with all equipment on that floor might require little or no protection. A simple way by which the hazard of spread of fire and smoke through a building may be effectively reduced is to install one or more small systems on each floor in place of large systems serving several floors. In this way passage of ducts through floors is eliminated, less space is taken up by ducts, and close control of atmospheric conditions in the various sections of the building can be more easily obtained. In existing buildings this method may sometimes result in less expensive installation as well as better control of fire and smoke hazards and should have the serious consideration of air conditioning engineers.

The proper installation and maintenance of an air conditioning system should include most, if not all, of the following features:

1. Fan control conveniently located.
2. Incombustible ducts, linings, and laggings.
3. Adequate cleanout and inspection openings in duct system.
4. Small mesh screen or grille in return duct openings.
5. Fresh air intake well away from exposures.
6. Incombustible filters changed or cleaned periodically.
7. An adequate refrigeration code.
8. Fire dampers to cut off parts of system.
9. Proper fire protection.
10. Careful and periodic inspection.
11. Cleanliness and good housekeeping.

The regulations require fire doors and dampers in the ducts under certain conditions. Fig. 3 shows a fire door installation in a duct and is required where the duct

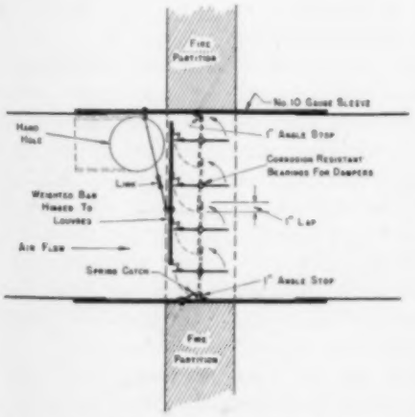


Fig. 2—Damper arrangement in fire partition

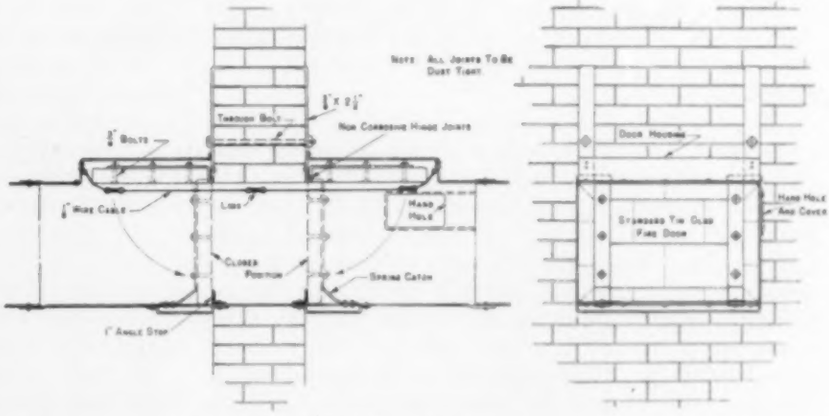


Fig. 3—Suggested type of automatic hinged fire doors for duct passing through opening in fire wall

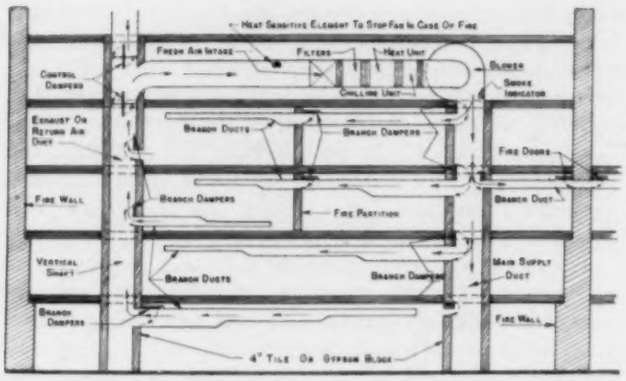


Fig. 4—Typical installation of an air conditioning system in a building of fireproof, semi-fireproof, or heavy mill construction

passes through an unpierced fire wall or a fire wall with openings protected by fire doors, while Fig. 2 shows a damper installation, required where floors are required to be separated. When discussing this requirement with members of the air conditioning industry, it was surprising to find that the general attitude seemed to be that this was an insurmountable problem which could not be complied with, until actual illustrations were presented and it was shown that the many installations might not require such cut-offs. Probably the simplest way in which this part of the regulations might be stated is that if under local construction requirements, openings are permitted through a partition, wall, or floor without protection at such openings, an unprotected duct opening would be permitted through the same partition, wall, or floor. On the other hand, if the local building code requires with respect to a certain type of construction and occupancy that an elevator or stairway be enclosed in a masonry shaft with all openings protected by fire doors, it naturally follows that a vertical air conditioning duct passing through the floor should be enclosed in masonry with a damper in the duct outlet at each floor. Fig. 4 shows such a typical installation.

The question of the extinguishment of a fire at the filters in an air conditioning system can readily be taken care of by automatic sprinklers or some other form of fire extinguishment, but if the system covers a large area and combustible material is permitted to collect in the horizontal ducts, it is impracticable, if not impossible, to protect the hazard. About the only solution to this problem is the periodic inspection and cleaning of the duct system at intervals of sufficient frequency so that should a fire occur, no combustible debris will be involved.

The present regulations of the *National Board of Fire Underwriters* require an approved manually-reset thermostatic device set at a maximum of 125 F, which is arranged to shut down the fans automatically when the temperature of the air in the system becomes excessive, as from fire. While there is such a device used principally in connection with domestic, forced draft, hot air heating systems, there is no device meeting these requirements which is now listed by Underwriters' Laboratories, Inc. The present regulations covering the protection of walls and partitions take care of the specifications covering fire cut-offs in ducts, but there is no reason why there will not be special designs of fire doors and dampers which will be submitted to the Laboratories for testing and listing. The fire resistive qualities of duct linings may also be investigated, as well as extinguishing agents of different types, as to their practical application with respect to air conditioning. Some form of dependable smoke detection will undoubtedly be investigated. At the present time, the only reference to such protection is contained in a note printed in the August, 1938, amendment to the rules, which reads as follows:

It is recognized that in certain occupancies there is a potential panic and smoke damage hazard. For such conditions it is recommended that consideration be given to the development and application of suitable smoke detectors to automatically shut down the system and cause an alarm signal.

Several years ago when air conditioning began to really develop, it was thought that the filter was the disturbing factor as far as the fire hazard was concerned. Then fires in combustible duct linings called attention to the fact that ducts should be of incombustible construction. But the latest development and the fire which occurred in a Detroit department store showed that still other factors such as combustible debris in the duct system might add to the panic and smoke loss. A new system can be so designed that this smoke hazard is reduced to a minimum but, in connection with those not so designed and old systems not in compliance with the rules, some form of smoke detection will undoubtedly be required which will operate at sufficient speed to prevent smoke and hot gas from being discharged into buildings housing occupancies where the panic hazard is pronounced. Although the protection of each air conditioning system is a special engineering problem as the designs generally differ in some respect, the purpose of the smoke detection system would be to shut down the fan and close dampers, preventing the smoke from entering the building and discharging it through a suitable opening, the fire being controlled by some extinguishing agent not connected with the smoke detecting apparatus.

Although the developments in the air conditioning industry are advancing rapidly, it can readily be seen that many of the necessary regulations are at present in a state of flux. The air conditioning profession and industry are cooperating in the development of suitable protection, for they realize that if acceptable safety features can be incorporated in these systems it will eliminate the possibility of seriously retarding the progress of the entire industry. Proper design and maintenance will materially assist in obtaining an acceptable installation which can be suitably protected.

Due to the inherent features of an air conditioning system, it is impracticable to suggest specific fire protection that would properly apply to all cases, and therefore it is necessary to give each system requiring protection special study and treatment. Most systems, and especially the smaller ones and those confined to one floor or part of a floor, will not usually require any special fire protection if installed according to the regulations, but the large systems supplying more than one floor from below may present a hazard which should be protected, regardless of installation methods. It then follows that the best assurance against loss by fire is not only a standard installation, but cleanliness at all times, good maintenance, and constant vigilance by the owner or occupant

Pittsburgh Experiment Station of the U. S. Bureau of Mines where the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS is located

Summer Cooling Requirements in Washington, D. C., and Other Metropolitan Districts

By F. C. Houghten*, (*MEMBER*), Carl Gutberlet** and Albert A. Rosenberg***, Pittsburgh, Pa.

SINCE 1935 the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has been studying the cooling requirements of persons seated at rest or engaged in sedentary occupations in summer cooled and air conditioned spaces. During the summers of 1935 and 1936, laboratory studies of trained subjects were made by research men in Pittsburgh, Toronto, and College Station, Texas. The findings in these studies[1, 2] showed that for persons in any geographical region, the effective temperature scale applied without any very pronounced independent effects of relative humidity over a fairly wide range. Little variation in the indoor summer cooling requirements was found for Pittsburgh and Texas, having maximum mean normal summer temperatures of 74.5 and 84.0 F, respectively. However, for Toronto, having a considerably lower maximum mean normal temperature of 68.2 F, the subjects showed a desire for approximately 2½ deg lower effective temperature.

During the summer of 1937 laboratory studies were continued at College Station, Texas, and Toronto, Canada, while a fairly comprehensive field study was made of the requirements of 275 office workers in a summer cooled and air conditioned space in Minneapolis, Minnesota. The Texas and Toronto studies[3] more or less verified the findings of the previous year.

The Minneapolis study,[3] besides being very conclusive in regard to the requirements of the office workers in that city, indicated some variation in the requirements of different geographical regions. As a result, three studies similar to that made in Minneapolis during 1937 were carried out during the summer of 1938: The results of one of these studies, in San Antonio, Texas, have already been published;[4] the data from a second study, conducted in the air conditioned offices of the Metropolitan Life Insurance Building, New York, have not yet been com-

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[1]ASHVE Research Report No. 1035, Comfort Standards for Summer Air Conditioning, by F. C. Houghten and Carl Gutberlet. (ASHVE TRANSACTIONS, Vol. 42, 1936, p. 215.)

[2]ASHVE Research Report No. 1055, Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecke, C. Tasker and Carl Gutberlet. (ASHVE TRANSACTIONS, Vol. 43, 1937, p. 145.)

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Mackinac Island, Mich., July 1939.

[3]ASHVE Research Report, Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley. (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, Dec. 1937, p. 758.)

[4]ASHVE Research Report, Reactions of Office Workers to Air Conditioning in South Texas, by A. J. Rummel, F. E. Giesecke, W. H. Badgett and A. T. Moses. (Ibid, May 1939, p. 323.)

pletely analyzed and will be reported at a later date. This present report deals specifically with the results of the study conducted in the air conditioned offices of the Federal Reserve Building in Washington, D. C., with some comparison of the results of all the previous studies.

Through the cooperation of the building management a number of offices in the modern, air conditioned Federal Reserve Building in Washington were made available for the study from August 1 to August 31, 1938. These offices were typical of the rest of the offices in the building, and included 73 workers of both sexes, of wide age distribution and rank in the organization.

Throughout a study during any certain day a predetermined effective temperature was maintained through the operation of the central air conditioning system, while a trained observer recorded the atmospheric and personal data. The data collected included a record of outside dry-bulb and wet-bulb temperatures, wind velocity and other weather conditions. To obtain inside conditions, the observer carried a motor-driven psychrometer which he placed in a location construed to be representative of air conditions in that room or portion of the room. This instrument was allowed to adjust itself to equilibrium while the occupants were interrogated, and after the elapse of sufficient time the dry- and wet-bulb temperatures were recorded; the data of the room conditions and answers of the occupants were recorded on a form similar to Table 1.

Previous to the actual study, the 73 workers participating were given case numbers, and their individual characteristics, including age or approximate age, stature, weight, apparent health, etc., were recorded for future reference. As shown in Table 1, the following observations were made: Time of day at which the observation was made; the dry-bulb and wet-bulb temperatures representative of air conditions in which the worker was present; the worker's reaction upon leaving the cooled space that afternoon (recorded the day following); his reaction to entering the cooled space that morning; his current feeling of warmth and degree of perspiration; and the classification of his clothing as to light, medium, or heavy weight. From the dry-bulb and wet-bulb temperature readings the relative humidity and the effective temperature were later determined and recorded.

The same index of feeling of warmth employed in earlier studies was used. According to this index, a feeling of ideal comfort is recorded as *C* or given the number "4"; comfortably cool and comfortably warm are indicated as *MC* and *MW*, and numbered "3" and "5", respectively; feelings of severely cool and severely warm are indicated as *SC* and *SW*, and numbered "2" and "6", respectively; while decidedly cold and decidedly hot are numbered 1 and 7, respectively. In this study, however, only "2" to "6", inclusive, were used.

Degree of perspiration was indicated in accordance with the index used in earlier laboratory studies; that is, "0" for forehead or body, dry; "1" for forehead or body, clammy; "2" for forehead or body, damp (perspiration just visible); "3" for forehead or body, wet (sweat covering the surface, frequently in drops); "4" for perspiration on the forehead running down, or perspiration on the body running down and wetting the clothing. In this study, however, most of the observations were taken after at least one hour's occupancy in the cooled space, and for this condition there was usually no perspiration, which is indicated by the letter *N*.

The data were collected by employees of the Federal Reserve Board and were sent to the Research Laboratory of the Society for analysis. In analyzing the data, all observations made on all workers present on all days during the study were tabulated according to the effective temperature pertaining with relation to the comfort index. The total number and, from that, the percentage of the total number of observations for each comfort index at each degree effective temperature was tabulated; from these tabulated data the curves in Figs. 1 and 2 were plotted. In these charts "ideal comfort" curves, or index "4", are plotted for all workers taking part in the study, all workers over 40 years of age, all workers under 40 years of age, all men, and all women. It will be observed that in each case a maximum-distribution curve is indicated, showing a maximum percentage of comfort at some definite effective tem-

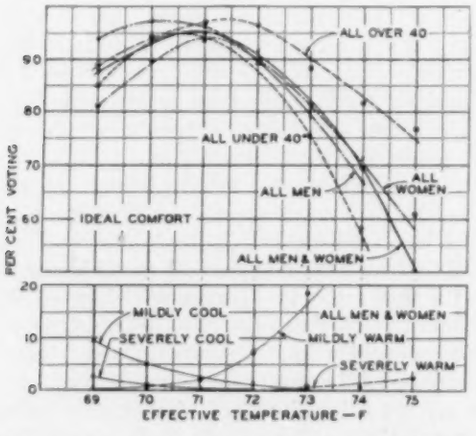


Fig. 1—Relation between effective temperature and the per cent of office workers in Washington indicating ideal comfort. Curves for all men and women, all men, all women, all over 40, and all under 40. Percentage curves in lower chart indicating other feelings of warmth are for all men and women

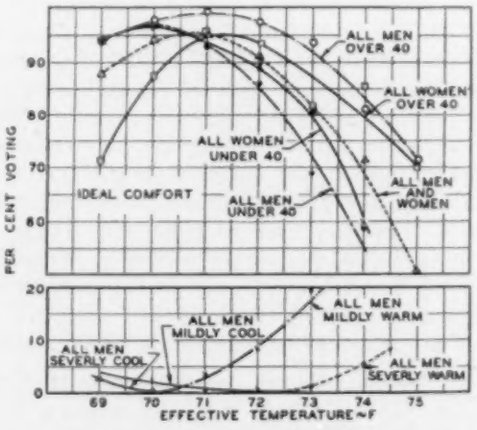


Fig. 2—Relation between effective temperature and the per cent of office workers in Washington indicating ideal comfort. Curves for all men and women, all men over 40, all men under 40, all women over 40 and all women under 40. Percentage curves in lower chart indicating other feelings of warmth are for all men and women

perature, with a falling off in percentage in either direction. In order not to subject the workers to undue extremes of low and high temperatures, most of the observations were limited to the effective temperature range of 69 to 75 deg; while a few observations were made below and above these effective temperatures, the number were too few to give accurate indications. Hence, only a small portion of ideally comfortable curves are drawn differing in this respect from some of the curves in the earlier reports. However, the main objective of this study was to determine the temperature range over which the maximum percentage experienced comfort, rather than a study of conditions where the percentage of comfort was small and the percentage of discomfort was great.

In general, the men indicated a maximum percentage feeling comfortable at a slightly lower temperature than women; while workers over 40 years of age, either for the combined group or for either sex, show an optimum percentage of comfort at a slightly higher temperature. One characteristic difference in the results of this study from the one made in Minneapolis during 1937 is the high percentage indicating "ideal comfort" for any group. This is undoubtedly a result of the difference in method of collecting the data. In the Minneapolis study, daily questionnaire cards were given to the individual workers to fill out, while in the Washington study a trained observer obtained and recorded the information. This resulted in more consistent replies for this study; and also, the direct observation of the wet- and dry-bulb temperatures in the immediate vicinity of the worker probably gave a better indication of the true atmospheric condition.

Table 1—Daily Report Record

Thursday, August 4, 1938 Sheet No. 1

CASE NO	TIME	INSIDE AIR CONDITIONS WB	DB	ET	RH	PM LEAVING REACTION	AM ENTERING REACTION	CURRENT FEELING S.C – M.C C MW – S.W	PERSPIRATION FOREHEAD	CLOTHING	AGE	SEX	OUTDOOR CONDITIONS TIME OF ENTRY 9 AM DB	WB	TIME OF LEAVING 3 PM DB	WB
13	9:52 2:00	63 64	75 75	70.1 70.5	52 56	NO	NO	C C	N	L	25	F	84	77.5	87	79
14	9:52 2:00	63 64	75 75	70.1 70.5	52 56	NO	NO	C C	N	L	45	F	84	77.5	87	79
15	9:54	63	75	70.1	52	NO	NO	C	N	L	29	M	84	77.5	87	79
16	9:56	64	77	71.5	50	YES	NO	C	N	L	44	F	84	77.5	87	79
17	9:58 2:03	64 63	79 77	72.3 71.2	44 46	NO	NO	MW C	N	L	30	M	84	77.5	87	79
18	10:00 2:05	64 64	79 78	72.5 72.0	44 47	YES	NO	C C	N	L	41	F	84	77.5	87	79

The expressions of feeling of comfort "2" and "3" and "5" and "6" are also plotted in Figs. 1 and 2, for all men and women and for all men, respectively. The same general characteristics for these curves are indicated as were found in the earlier studies; that is, a minimum of discomfort results at approximately the same effective temperature as maximum comfort, with a slightly higher percentage indicating mildly cool than those indicating mildly warm for the condition of optimum comfort.

Data on the shock experienced by the workers upon entering the cooled space in the morning and the reaction upon returning to the hot outside during the afternoon were collected. While there were a few indications of shock or reaction, the number never exceeded 2 per cent, and this small percentage showed no consistent relationship to either the effective temperature maintained indoors or the outside temperature before entering or after leaving. This study therefore adds nothing to this phase of the subject beyond the conclusions resulting from the previous Minneapolis study.[5]

It is of interest to compare the results of this study

[5]ASHVE Research Report, Shock Experiences of 275 Workers After Entering and Leaving Cooled and Air Conditioned Offices, by A. B. Newton, F. C. Houghten, Carl Gutberlet, R. W. Qualley, and M. C. W. Tomlinson. (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, July 1938, p. 481.)

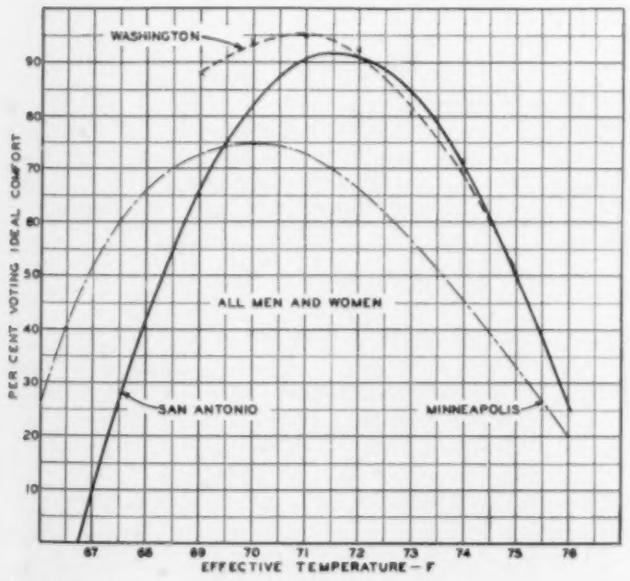


Fig. 3—Relation between effective temperature and the percentage of persons indicating ideal comfort. Curves for all men and women in Washington, San Antonio and Minneapolis

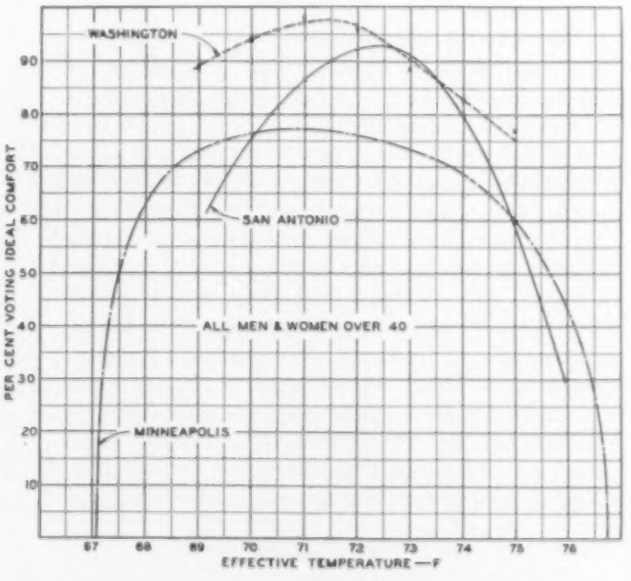


Fig. 4—Relation between effective temperature and the percentage of persons indicating ideal comfort. Curves for all men and women over 40 in Washington, San Antonio and Minneapolis

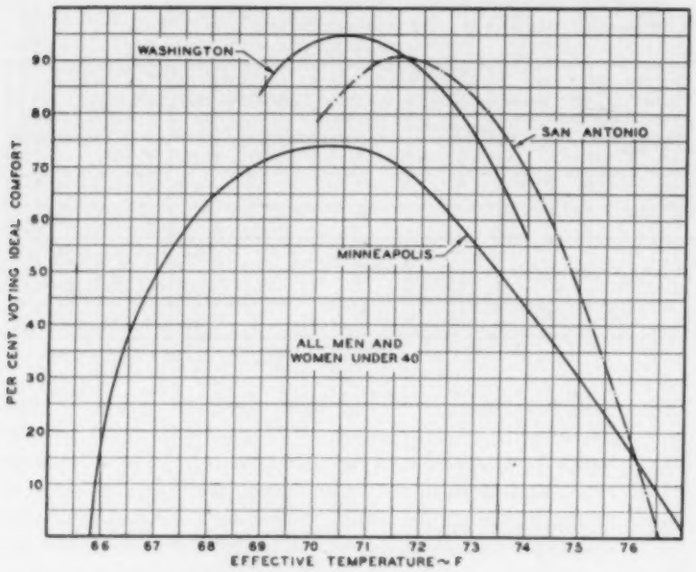


Fig. 5—Relation between effective temperature and the percentage of persons indicating ideal comfort. Curves for all men and women under 40 in Washington, San Antonio and Minneapolis

with the other studies by or in cooperation with the Society's Laboratory since 1935, and this is done for the different worker or subject groups in Figs. 3 to 7, inclusive. Curves of "ideal comfort" are drawn for each metropolitan district from which data were available, but in comparing these curves it should be emphasized that the maximum percentage indicating ideal comfort in any locality is probably affected by the method of collecting the data more than from the difference in actual feeling of comfort. The effective temperature at which optimum comfort occurs, however, should be largely due to the difference in geographical regions, possibly with some variations due to lack of or imperfection in imparting to the persons interrogating, the truth of what should be construed as ideally comfortable. However, it is probable that this factor is not of great importance. The curves in these charts show a lower temperature as desired by the Minneapolis and Toronto workers or subjects, with relatively little variation due to geographical location in other instances, where the same type of individual is concerned.

A review of the data plotted in Figs. 1 to 7, inclusive, gives a fairly concrete picture of cooling requirements for summer air conditioning for widely scattered metropolitan districts. The data collected in Washington last summer and plotted in Figs. 1 and 2 indicate that about 95 per cent of all workers in the Federal Reserve Building during the period from August 1 to August 31 were comfortable at 71 ET. If a tolerance of plus or minus one degree is allowable a minimum of 91 per cent was comfortable. These facts are of much greater practical significance than the small variations in effective temperature for the maximum percentage of comfort shown for different age and sex groups. As an example, compared with about 95 per cent comfortable at 71 ET for all groups, about 97 per cent of those over 40 years of age were comfortable at 71½ ET, with about the same percentage of comfort for all men at a little above 70 ET, and for all men and women under 40 the percentage was 95 at 70½ ET. While these differences are of some significance when considered from an academic point of view they cannot be given much consideration in practical application of air conditioning. Further, it is of interest to note that the optimum percentage for all curves falls well within the range of 70 to 72 ET, again indicating that a practice of maintaining 71 ET with a tolerance of plus or minus one degree would satisfy all groups. If an operating tolerance of 2 deg were required it would reduce the percentage of comfort of any group to approximately 70 per cent. It is also of interest to note that the minimum percentage of those feeling either mildly or slightly warm or cold also comes within this temperature range.

When viewing the results of the several studies in different metropolitan districts, one is confronted with a greater variation. Inspection of the curves shows that they more or less fall into two groups, the Minneapolis and Toronto data showing consistently a desire for lower temperatures than those for the other cities. All curves for Minneapolis and Toronto show maximum percentages at about 70 ET with variations in Minneapolis for all women, and all men and women over 40 of 70½ and 71 ET, respectively. Curves for the metropolitan districts other than Toronto and Minneapolis show maximum percentages at effective temperatures ranging from about 70½ for all men and women under 40 in Washington, to 73 deg for all men in the College Station, Texas, study. However, it should be pointed out that the broken-line curves in Fig. 6 for Pittsburgh and College Station, Texas, (this was also the case for the Toronto

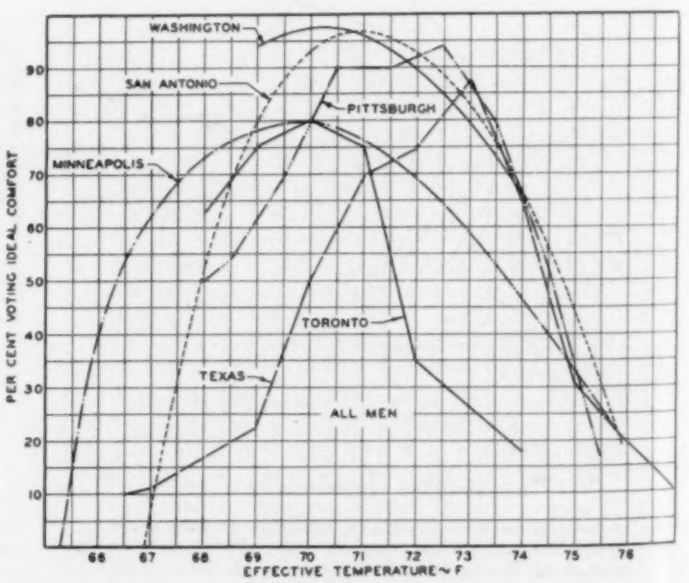


Fig. 6—Relation between effective temperature and the percentage of persons indicating ideal comfort. Curves for all men in Washington, San Antonio, Minneapolis, Pittsburgh, Toronto and Texas

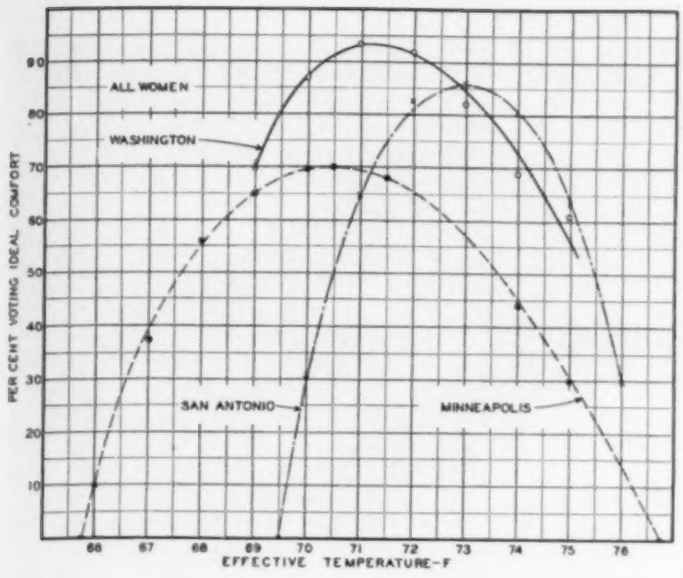


Fig. 7—Relation between effective temperature and the percentage of persons indicating ideal comfort. Curves for all women in Washington, San Antonio and Minneapolis

study) resulted from studies on relatively few laboratory subjects, giving insufficient data for a statistical analysis resulting in smooth curves. Judging from the nature of the curves it is probable that statistical data for Pittsburgh and College Station, Texas, would have given maximum percentages voting comfortable at about 71½ and 72½ ET, respectively, which would reduce the effective temperature spread to from 70½ to 72½ deg, or a total spread of 2 deg.

In this connection, if one were to accept 71½ deg as the best single practical effective temperature to have been maintained in all of these instances, it would have given percentages of comfort of about 72 for all men at College Station, Texas, 90 for all men at Pittsburgh, 95 for all men at San Antonio, and 93 for all men at Washington; 75 for all women at San Antonio and 93 for all women at Washington; 90 for all men and women over 40 at San Antonio, and 97 for all men and women over 40 at Washington; 91 for all men and women under 40 at San Antonio and 92 for all men and women under 40 at Washington; 92 for all men and women at San Antonio, and 94 for all men and women at Washington;—compared with maximum percentages for the same groups and locations of: 87, 94, 97, 97, 86, 93, 93, 97, 91, 95, 92, 95, respectively. With the exception of the men studied at College Station, Texas, and the women studied at San Antonio, the percentages which would be adversely affected by maintaining a common condition of 71½ ET in all locations would be insignificant.

It is of interest to compare these deductions with the normal mean summer temperatures for some of these localities. Fig. 8, taken from an earlier Laboratory publication,[2] gives such data for Boston, Toronto, Pittsburgh, and College Station, Texas. It will be noted that the mean summer temperature in Toronto is considerably lower than those for Pittsburgh and San Antonio, Texas.

It is probable that the normal mean temperature in Minneapolis would fall more nearly around that of Toronto, while that of Washington would probably fall between that of Pittsburgh and San Antonio. This might indicate that localities having a maximum normal mean summer temperature of 74 F or above should maintain an effective temperature of about 71 deg, while those localities showing normal mean summer temperatures of below 74 F should maintain effective temperatures below 71 deg. Beyond this, some adjustment, probably not to exceed one degree effective temperature, might be based upon peculiar local conditions or peculiarities of the type of occupants.

It should be emphasized that the data contained in this report and the deductions offered previously apply to persons entering the cooled space and remaining for three hours or more, which may be construed as permanent occupancy; that is, the occupants remain in the cooled space long enough to become in equilibrium with it. There are reasons for questioning the application of these results to occupancy of a shorter period. The shock data published in earlier reports[2, 4, 5] indicated that maintaining the suggested effective temperature results in no serious shock, and it should be emphasized that this same shock will be experienced whether a person enters the cooled space for 5 min or for permanent occupancy. Hence, the same conditions as are maintained for permanent occupancy should give satisfactory results for occupancy of shorter duration. However, for shorter periods of occupancy the people may be reasonably satisfied with the sensation of coolness acquired during the short period of time. Therefore, it may be reasoned with some justification that while for a short period of occupancy maximum comfort may be expected in the same conditions as are maintained for permanent occupancy, the occupants may be as well satisfied in a condition of higher effective temperature, which would of course be more economical to maintain.

Acknowledgment

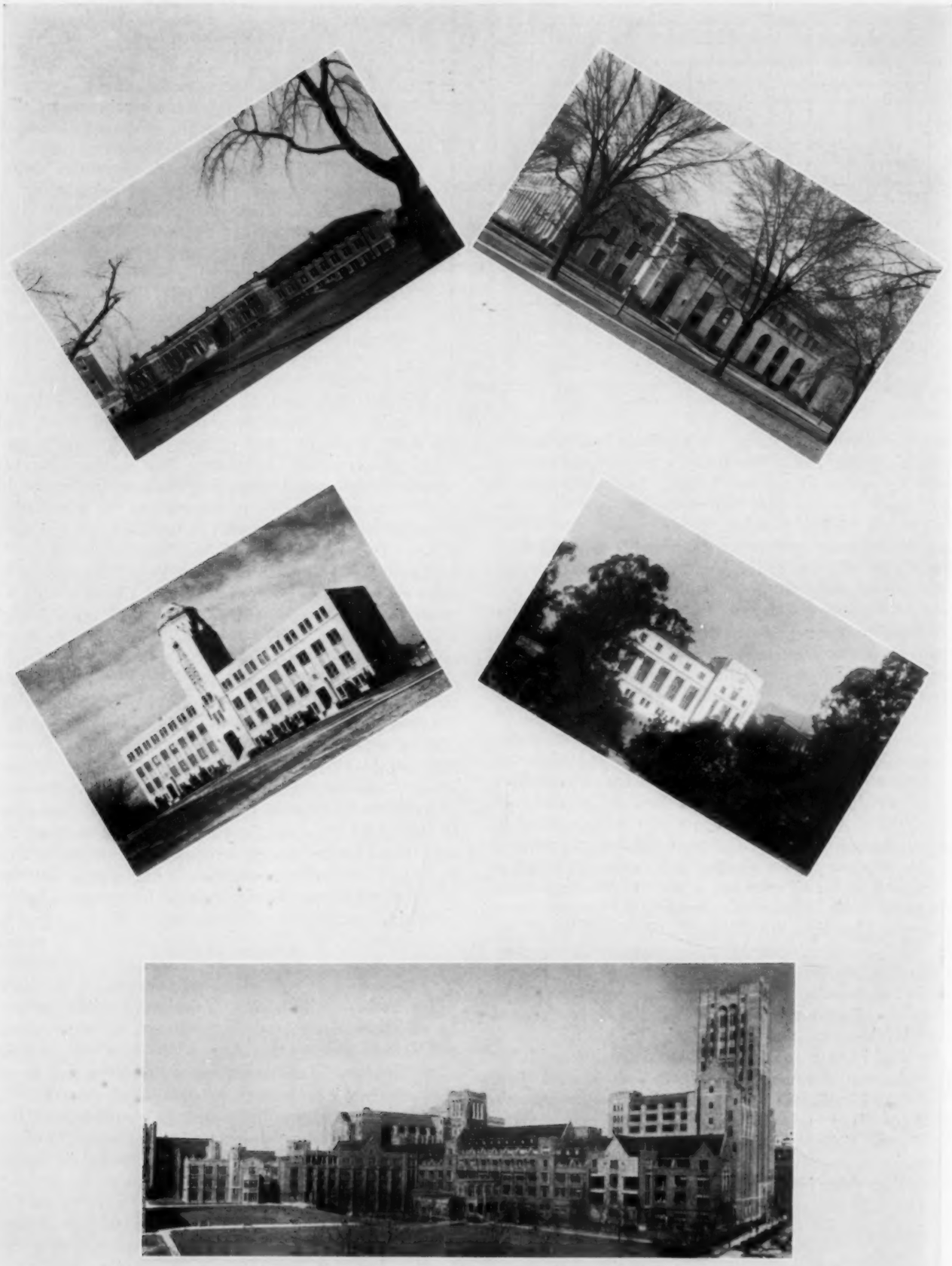
The authors acknowledge the valued services rendered by members of the staff of the Board of Governors of the Federal Reserve System, and its building manager, F. E. Spurney, in making the study possible, and for all their services in collecting and tabulating the data.

The authors also acknowledge the valuable assistance rendered by Thomas Urdahl and other members of the Washington, D. C., Chapter of the ASHVE in making arrangements for the investigation.

New ASA Standard

The *American Standards Association* recently announced approval of a new standard for dimensions and materials of wrought-iron and wrought-steel pipe and tubing which should result in considerable savings in cost and convenience to manufacturer and user industries.

This new standard consists of ten weight and thickness schedules running from the lightest to the heaviest pipe commonly used. Tabular values of weights for corresponding sizes of steel and wrought-iron pipe are given and the nominal wall thicknesses have been adjusted for the difference in specific weights.



Top left—University of Minnesota, Minneapolis . . . Top right—University of Wisconsin, Madison . . . Center left—Agricultural and Mechanical College of Texas, College Station . . . Center right—University of California, Berkeley . . . Bottom—University of Illinois Medical School, Chicago

Cooperative Research Sponsored by Society at Various Institutions

SINCE the inception of the ASHVE Research Laboratory in 1919 the Committee on Research has actively sponsored cooperative investigations in a number of colleges, universities and technical schools. This arrangement has proved especially effective in broadening the work undertaken and at the same time enabling the available budget to accomplish maximum results with minimum costs, by avoiding the duplication of existing equipment and qualified personnel.

At first this phase of the Society research program was necessarily restricted and not until 1927 were funds available in sufficient amount to accomplish outstanding results which have received world-wide recognition and acclaim. This research has been so beneficial to the Society that increasing amounts have been allocated each year for this purpose and in 1939 almost $10,000 will be expended towards cooperative agreements with funds provided from membership dues allocated to research and through earmarked funds contributed from various sources. This year research is being conducted in 11 institutions on a variety of subjects and the following brief outline will indicate the general extent of the work being investigated.

University of California

Four types of forced draft and atmospheric cooling towers have been placed on the roof of one of the engineering buildings in such a manner that the equipment can be rotated by means of a specially constructed platform. Data are being collected to establish a minimum rate of evaporation which will occur from a stationary, horizontal or vertical water surface into air. Computations will establish the rate of fall for water drops of different diameters and from these results cooling rates may be predicted in any tower provided the drop size distribution is known. Several effectiveness ratios of cooling towers have been proposed based on the concept of an air heating device and of a water cooling device, each of which has its place in design calculations and this is another phase of this particular subject which will be investigated in the near future.

Case School of Applied Science

Three projects are under investigation at this school which are of tremendous interest to air conditioning engineers. One deals with the behavior of air streams and jets discharging into free open spaces wherein special attention is being given to the angle of the stream and the velocity distribution pattern. Preliminary studies have been made on the methods of calibration and the use of seven different instruments for ascertaining air velocity measurements. These methods are being compared in five different kinds of air streams involving a wind tunnel, discharge from a plenum, discharge from a long duct, discharge from a nozzle and on the end of a rotating arm. A third project being studied deals with the heat transfer of finned tubes by forced air circulation. Results of several years of research at this school are being studied and correlated and a few additional tests are being conducted to determine the effect of refrigeration temperature on the performance of dehumidifying coils. The entire program is being directed towards the formulation of uniform practices which eventually can be used as a basis for developing a standard Society code for rating and selecting dehumidifying coils.

University of Illinois

Engineering School

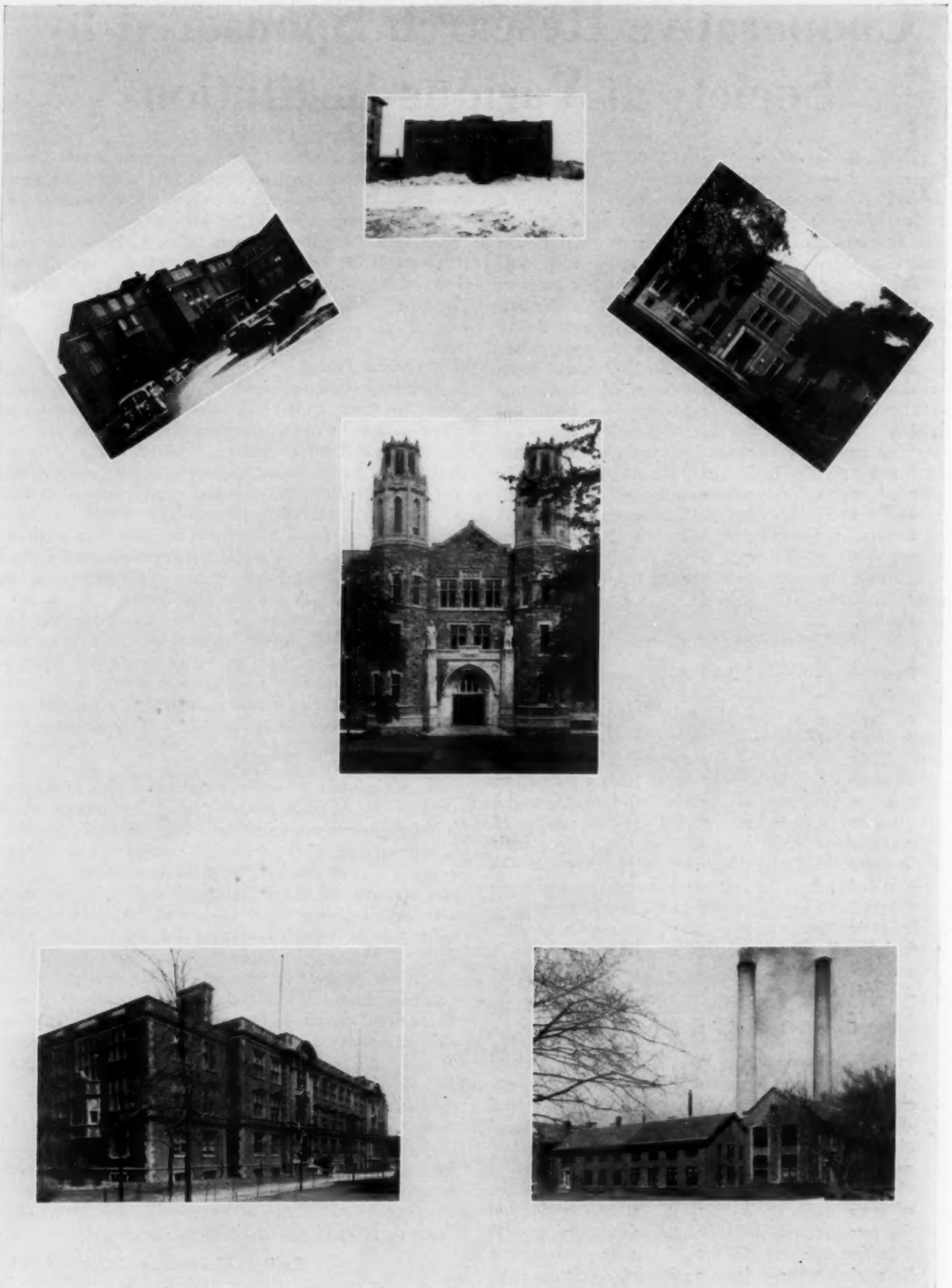
The work of this cooperative investigation is concerned with the effect of inlet and exhaust opening location on the motion and distribution of air within a room. The test room in which these studies are being conducted is located in a larger insulated room provided with direct expansion refrigeration coils and electric heaters so that control temperatures may be maintained around the outside of the test room. Provision has also been made so that two or three walls of the test room may be exposed to these conditions. Present plans are to determine air distribution with various inlet air velocities ranging from 300 to 1200 fpm when cooling the room under varying conditions. Work during the first part of 1939 on this project has been confined principally to the construction of suitable anemometers for measuring the air velocity at a large number of locations in the test room and construction of necessary apparatus for their accurate calibration.

Medical School

An air conditioned room has been provided in the hospital of the Medical School located in Chicago where healthy subjects are being studied under basal conditions in atmospheres within the range of the Society comfort zone. Observations have been made on the cardiac output, quantity of blood flowing through the periphery and the circulating blood volume of the subjects. Results thus far reported indicate that the cardiac output does not rise on the exposure of a resting man to hot conditions until a corresponding rise in heat production has been noted. The blood flow through the periphery, particularly the hands, becomes very great as the mean skin temperature approaches the rectal temperature. Also it was found that the blood volume rises within periods of two or three hours exposure to hot wet conditions. Present experimental work involves a direct continuance of these studies and in addition to the observations previously noted data are now being collected on the skin temperature, oxygen saturation of venous blood, pulse rate, respiration rate, vital capacity, blood pressure and rate of loss of insensible perspiration from the several test subjects.

Lehigh University

Negotiations were consummated early this year for work to be done at this University covering studies on the frictional resistance to the flow of air in straight



Top left—University of Pittsburgh, Magee Hospital . . . Top center—Michigan College of Mining and Technology, Houghton . . . Top right—Case School of Applied Science, Cleveland . . . Center—Lehigh University, Bethlehem . . . Bottom left—University of Pennsylvania, Philadelphia . . . Bottom right—University of Illinois Engineering Laboratory, Urbana

ducts and elbows. Round elbows tested in the ASHVE Research Laboratory in Pittsburgh will be sent to Lehigh for duplicate tests to check and confirm the results obtained in the Society Laboratory. Equivalent square and round elbows with different radii will be studied and tests will be repeated on all these elbows with turning vanes to determine the resistance of the flow as well as distribution.

Michigan College of Mining and Technology

Corrosion studies are being continued this year with special emphasis being placed on a single phase of the problem encountered in steam heating systems dealing with the deterioration of pipe. The determination of the amount of non-condensable gases which will dissolve in the condensate formed in a heating system at different rates of condensation when the steam contains a known amount of these condensable gases is the phase of this problem which has been selected for closer study with the equipment available. Corrosion rates will be studied but this part of the problem will be considered of secondary importance to the contamination difficulties. Tests will be run with different sources of supply water such as city, school and possibly distilled water artificially contaminated to the degree desired.

University of Minnesota

Investigation of air filter performance and methods of rating air filters has been under way at this institution for a number of years. Different types of dust are being studied for the purpose of selecting a standard which will have uniform properties and may be duplicated. Several types of lamp black and carbon black are under investigation and the tentative results indicate that certain brands of lamp black or carbon black are much more uniform in quality and therefore could be duplicated with reasonable certainty. The effects of air temperatures on the performance characteristics of different types of air filters are being tested with variations in air temperatures from 0 to 130 F. Dust particle size and density have been found to have an effect on filter performance and the problem remains to select a dust which could be duplicated with certainty and which will be as near as possible to the average dust found in the air. A suitable field test is also being studied which can be correlated with the standard Society rating test code and which may be practical for checking the performance of filters in actual installations.

University of Pennsylvania

This research contemplates experimental investigations to determine the departures from Dalton's Law in the case of moist air with the ultimate objective being a formulation of psychrometric data having a legitimate claim to universal acceptance as a standard on account of thermodynamic consistency. Statistical mechanics gives the form of departure from Dalton's Law but leaves undetermined a certain temperature function called the interaction constant expressing the mutual effect between the water vapor and dry air. This interaction constant is being determined experimentally from apparatus constructed with the idea of weighing the air with all practicable precision. Opinions are held that the results of this study will indicate that the saturation curve on a psychrometric chart drawn from application of Dalton's Law will be shifted both horizontally and vertically and the isotherms which under Dalton's Law are straight will be given some curvature.

University of Pittsburgh

This cooperative work is being conducted at the Elizabeth Steel Magee Hospital which has operating rooms and recovery wards equipped with air conditioning apparatus. Several devices are available for cleaning the air which consist of a cloth filter, electrostatic precipitator and air washer. Present studies are intended to determine whether these various systems of air cleaning actually decrease the bacteriological content of the air delivered to an operating room and whether this cleaned air actually decreases the bacteriological content of the air during operations when the room contains a number of people. Two methods of taking samples are being used, one involving blood agar pour plates to determine the amount of settling bacteria and the other will involve air samples taken by the Wells air centrifuge. An ultra-violet sterile lamp is also being used as a final method for purification to determine its actual value in addition to the other systems of air purification.

Agricultural and Mechanical College of Texas

This research has been planned and is being conducted to ascertain the flow of heat through walls for both heating and cooling cycles. To determine the heat absorbed by a wall it is necessary to know the heat conductivity and capacity of materials making up the wall construction and this study is intended to produce more accurate methods of determining the heating and cooling loads of buildings. To determine the conductivity and capacity of a wall two methods are being used, one being a small cubicle built in the laboratory of which one wall is composed of brick and hollow tile and equipped with thermocouples and heat flow meters so that the flow of heat through the construction can be studied carefully. In the other method resistance wires have been installed in a brick wall of one of the college buildings and in this instance the flow of heat is also being studied and the results tabulated.

University of Wisconsin

This study is intended to determine the effect of duct approach on the air discharge from a supply grille. The problem involves the determination of the velocities and distribution pattern of each part of the air stream issuing from the opening with the objective directed towards devising a method of producing equal flow over the entire grille opening and at the same time maintaining the frictional resistance at a desirable level. A testing apparatus has been arranged so that various stack and elbow designs may be attached to the top of a plenum chamber which is being provided with measured quantities of air from a motor driven fan. For determining direction of air leaving the opening an angle measuring device made of aluminum foil is being utilized which consists of a piece of metal counterbalanced and mounted on a needle shaft and provided with a suitable scale arranged so that the direction of air flow can be measured when the instrument is held at the edge of the stack under observation.

CORRELATING THERMAL RESEARCH

As a part of the efforts of the ASHVE Research Laboratory to correlate research in thermal engineering carried on by the many institutions engaged in such work, and to disseminate the published results of such studies together with other reports of progress in the field, and in order to make this information available to the membership of the Society, there is published monthly on this page a limited number of brief abstracts of articles which it is believed will be of interest to all concerned. For more complete lists address the Librarian, ASHVE Research Laboratory, U. S. Bureau of Mines Experiment Station, Pittsburgh, Pa.

F. C. Houghten, *Director.*

W. L. Fleisher, *Chairman*
COMMITTEE ON RESEARCH

A Simple Device for Sampling Air-Borne Bacteria, by Alexander Hollaender and J. M. Dalla Valle. *Public Health Reports*, U. S. *Public Health Service*, Vol. 54, No. 14, April 7, 1939, pp. 574-577. Description of a new air sampling device whereby the volume of air sampled can be accurately measured. Diagram of sampling device. The impingement principle is used; also the standard Petri dish method of culturing bacteria.

•

The Balancing of Compound Hot-Water Circuits, by Thomas H. F. Holman. *The Heating and Ventilating Engineer* (London), Vol. XII, No. 143, May 1939, pp. 509-513. Discussion of design of a hot-water heating system so that two or more companion circuits will have certain predetermined temperatures in the various portions of the apparatus, while in operation.

•

Boiler Ratings for Direct and Indirect Hot Water Supply Systems, by Henry H. Bruce. *The Heating and Ventilating Engineer* (London), Vol. XII, No. 143, May 1939, pp. 492-496. Discussion of boiler ratings for heating and for hot water supply systems. Inadequate storage supply capacity has caused boiler manufacturers to rate hot water supply boilers higher than heating boilers. This is unsatisfactory and should be changed by the designers of hot water supply systems.

•

The Possibilities of Heating by the Thermo-Pump System, by M. Egli. *Assoc. Suisse Elect.*, Bull., Vol. 30, pp. 42-46; Discussion pp. 46-48, Jan. 20, 1939. In German. The author gives a brief description of a thermo-pump system of heating followed by a treatment of the application of this principle to domestic heating. Comparisons are made with the cost of electrical heating systems.

•

Ventilation of Boiler Houses, by H. Schlicke. *Warme*, Vol. 62, pp. 25-27, Jan. 14, 1939. In German. The general problem of ventilation in boiler houses is discussed and this is shown to be a factor of sufficient importance to justify consideration when planning new buildings.

•

How to Select and Install Industrial Thermometers, by E. H. Hammond. *Power*, Vol. 83, No. 5, May 1939, pp. 88-89. Discussion of time-lag in a thermometer, method of determining time-lag curve for thermometer, and its use in determining the correct temperature when temperature to be determined is changing. Information which will help the manufacturer build the proper thermometer to order.

•

Selection of Direct Expansion Coils for Comfort Conditioning, by N. S. Shataloff. *Ice and Refrigeration*, Vol. 96, No. 6, June 1939, pp. 467-468. Discussion of variables involved in the selection of surface cooling coils. Description of the "Try and Check" method from the standpoint of the selection engineer, to enable him to make a rapid selection considering all the factors affecting performance of coils.

•

Nomographic Chart for Flow Analysis, by H. Shand, *The Canadian Engineer*, Vol. 76, No. 22, May 30, 1939, pp. 4-6. Chart compiled to facilitate the application of the Hardy Cross method of determining the flow distribution and loss of head in a water-supply network. Values can be read directly from chart.

•

A History of Science, Technology and Philosophy, in the Eighteenth Century, by A. Wolf. Published by The Macmillan Co., New York City. 798 pages and index. Advances were made in almost every field of intellectual enterprise and there was an unprecedented spread of knowledge beyond the circle of specialists. The story of each separate science is told and then that of technology in all its main branches.

•

A New-Type Orchard Heater. Combustion-Chamber Bowl-Type Orchard Heater Utilizes Inert Diluents, by A. S. Leonard. *Mechanical Engineering*, Vol. 61, No. 6, June 1939, pp. 443-446. Description of a new type orchard heater so designed that part of the flue gases returned to the fuel chamber is mixed with the incoming air, causing a reduced amount of soot in the bowl and of smoke output from the heater. This heater has a much wider range of burning rates with smokeless operation.

•

Unit and Duct Layout Gives Flexibility to Industrial Plant System, by Robert Grahn. *Heating and Ventilating*, Vol. 36, No. 5, May 1939, pp. 32-34. Description of the heating system in a new eight story building. Fresh air is drawn into the building at one point on each floor; it passes through a set of air filters, through two large heating coils and is distributed throughout the floor by usual ventilating system of ceiling ducts. Partly recirculated air is used.

•

British Degree-Days. *The Heating and Ventilating Engineer*, (London), Vol. XII, No. 143, May 1939, pp. 505-509. Discussion of British degree days and relation of steam consumption in relation to degree days. British degree days are based on 60 F which is 5 F lower than established indoor temperature, the same as U. S. degree-days based on 65 F is 5 F lower than the 70 deg standard indoor temperature. Table of degree-days for London for years 1934-1937.

•

Special Design Cuts Heating Cost. Builder Plans Homes and Builds to Reduce Equipment Charges. *Air Conditioning and Oil Heat*, Vol. XII, No. 6, June 1939, pp. 9-10 and 47. Description of 200 homes under construction near Arlington, Va., to be sold for less than $5000 each. Houses are semi-detached, of five or six rooms. Construction is brick with reinforced steel and concrete floors, all copper piping, round tile chimneys, and each is completely equipped with modern household utilities.

•

The Country School Goes Modern. *Domestic Engineering*, Vol. 153, No. 5, May 1939. Description of heating system in the Cool Spring School at Michigan City, Ind. The school has modern equipment for sanitation, ventilation, light and heat. The heating system consists of five boilers using forced hot water in a five zone control. Heat distribution is by a combination of unit heaters, reheaters and unit ventilators supplied with hot water from the forced circulation. The heating units are behind grilles where the heated air is mixed with washed filtered fresh air.

L. F. Kent
General Chairman

Marietta St., showing Henry Grady Monument

C. T. Baker
Vice-Chairman

Fall Meeting Down in Dixie

THE Atlanta Chapter will act as host for the first Fall Meeting of the Society, to be held at the Atlanta Biltmore Hotel, Atlanta, Ga., October 30 and 31. The Committee on Arrangements, headed by L. F. Kent, is making preparations for a big attendance of Society members, when Pres. J. F. McIntire calls the first session to order on October 30.

A schedule of three technical sessions has been planned and six papers are to be presented. Among the subjects offered for discussion are: a study of air measuring instruments, hospital air conditioning requirements, district heating, standard dust test, applications of air conditioning in food storage.

Members of the Atlanta Chapter are planning an entertainment program that will include a Get-together Luncheon, a Banquet and Dance, Special Inspection Trips and some entertaining events for ladies. The Committee on Arrangements is hard at work planning the program. The members of the committee are: L. F. Kent, General Chairman; C. T. Baker, Vice-Chairman; C. L. Templin, Reception; S. W. Boyd, Registration and Sessions; E. W. Klein, Banquet; T. T. Tucker, Ladies; A. H. Koch, Sports; G. H. Brodnax, Jr., Transportation; M. M. Crout, Publicity; and C. B. Cole, Finance.

Atlanta is the Capital of Georgia and is the leading industrial and commercial city of the Southeast. Rapid transportation is available from all parts of the country as Atlanta is served by eight main line railroads and it enjoys through train service from New York, Philadelphia, Washington, Jacksonville, New Orleans, Birmingham, Memphis, Cincinnati, Chicago, Cleveland, Detroit, Denver, Kansas City and St. Louis. Some typical schedules are:

Lv. Boston (New Haven)	8:00 a.m.	
New York (Penna. R.R.)	2:25 p.m.	(Penna. 11:30 a.m.
Philadelphia (Penna. R.R.)	4:07 p.m.	(Penna. 1:15 p.m.
Washington (So. Ry.)	7:00 p.m.	(Seaboard 4:00 p.m.
Ar. Atlanta (So. Ry.)	8:35 a.m.	(Seaboard 6:20 a.m.
Lv. Chicago (Big 4)	1:45 p.m.	(Penna. 1:00 p.m.
Detroit (N.Y.C.)	11:35 a.m.	
Toledo (N.Y.C.)	1:05 p.m.	
Cleveland (N.Y.C.)	3:05 p.m.	
St. Louis (L.&N.)		4:00 p.m.
Cincinnati (So. Ry.)	10:00 p.m.	(L.&N. 9:00 p.m.
Louisville (L.&N.)		7:15 p.m.
Lexington (So. Ry)	10:50 p.m.	(L.&N. 9:15 p.m.
Ar. Atlanta (So. Ry.)	10:00 a.m.	(L.&N. 8:25 a.m.
Lv. Jacksonville (So. Ry.)	11:05 p.m.	
Birmingham (So. Ry.)		7:15 a.m.
New Orleans (L. & N.)	6:30 p.m.	10:15 p.m.
Ar. Atlanta	8:10 a.m.	11:50 a.m.

Some examples of round trip railway fares and one way lower berth Pullman rates from some chapter cities to Atlanta are as follows:

	R.R. Fare Round-Trip	Pullman-Lower One-Way
Detroit	$ 38.80	$ 5.80
Chicago	41.25	5.80
Buffalo	49.20	6.85
Philadelphia	39.90	6.05
Kansas City	40.75	6.30
Seattle	117.05	20.25
New York	45.35	6.85
Toronto	52.75	7.10
Cleveland	38.80	5.80
Montreal	68.10	9.20
Washington	31.95	5.25

In addition to being the largest railway center of the South, the State is the center of a network of fine highways and those who plan to drive will find much of scenic beauty en route to Atlanta whether they travel via the Shenandoah or through the Smokies and the Land of the Sky.

Atlanta is a mecca for tourists and its location makes it a natural gateway between the north and the west. Also the city is a noted manufacturing center and produces 1500 different commodities. Because of its altitude, 1050 ft, Atlanta enjoys a fine

Replica of Bobby Burns Cottage

climate without extremes of either heat or cold. Its location in the rich Piedmont section 864 miles from New York and 792 from Chicago makes it accessible to a large percentage of the Society membership.

The headquarters for the meeting will be the Atlanta Biltmore Hotel and the room rates are as follows:

Single rooms with bath	$3.00	$4.00	$5.00 per day
Double rooms with bath	$4.50	$5.50	$6.50 (2 persons)
Twin bed rooms with bath	$5.00	$6.00	$7.00 (2 persons)

As a feature of the special get-together luncheon on Monday, the speaker will be Dr. M. L. Brittain, president of Georgia Tech. On Monday evening at the banquet, the speaker will be O. B. Keeler, world famous sports commentator and writer, who will interview the renowned golfer, Bobby Jones, who is a successful Atlanta lawyer and a director of the First National Bank of that city, as well as a director of the firm A. G. Spalding Co., New York.

The Committee on Arrangements has promised those members who come early a fine football game on Saturday afternoon, October 28, and on Sunday plenty of time for golf at the Bobby Jones course. There will be two fine technical meetings on Monday and Tuesday and an entertainment program for ladies.

Atlanta is the site of a number of colleges and universities, including Georgia School of Technology, Emory University, Oglethorpe University, Georgia Military Academy, and a number of schools for women.

Of interest to Atlanta visitors are Wren's Nest, the home of Joel Chandler Harris, creator of Uncle Remus; the only exact replica of Bobby Burns Cottage in America; Stone Mountain, the largest solid body of exposed granite in the world on which is being carved a monument to the Southern Confederacy; the Henry W. Brady monument erected for Georgia's most brilliant orator and journalist; Fort McPherson, headquarters of the 4th Corps Area; the Federal Penitentiary; State Capitol; Bobby Jones Municipal Golf Course; Eastlake Country Club; and the Eugene Mitchell Home where the author of *Gone With the Wind* grew up.

Western Michigan Members Meet

May 25, 1939. The annual meeting of the Western Michigan Chapter was held at the Kalamazoo Country Club, Kalamazoo, Mich.

Those who received awards in the annual golf tournament were as follows: Low net (Chapter cup)—H. J. Metzger, Kalamazoo; blind bogey—C. A. Dennison, Lansing; most 4's—Dave Satin; most 5's—H. V. Muir—Kalamazoo; most 6's—P.O. Wierenga, Grand Rapids; most 7's—S. A. Schaeffer, Kalamazoo; high gross—S. Harrison, Lansing. In the archery contest B. M. Smith showed the greatest skill.

Dinner was served to 92 members and guests and later roll call was taken and Secy. W. G. Schlichting read the minutes of the April meeting, which were approved.

Tellers were appointed and a count of the ballots showed that the following officers were elected for the 1939-40 season to assume office following the Society's Summer Meeting:

President—B. F. McLouth
Vice-President—T. D. Stafford
Secretary—C. H. Pesterfield
Treasurer—C. H. Morton
Board of Governors—C. R. McConner, W. G. Schlichting, F. C. Warren.

After the business meeting two moving pictures and a dancing act were presented before adjournment at 11:00 p. m.

Oregon Chapter Elects Officers

July 27, 1939. The Oregon Chapter was called to order by Chairman B. W. Farnes who stated that the purpose of the meeting was to elect officers and approve By-Laws as the Council of the Society had approved the petition for a chapter charter.

E. E. Carroll, as chairman of the Nominating Committee, submitted a report containing the following nominations:

For President—J. D. Kroeker
For Vice-President—T. E. Taylor
For Secretary—C. E. Heinkel
For Treasurer—H. B. Nielsen.

Mr. Farnes called for further nominations and, since none were offered, C. E. Enders moved that the nominations be closed. This was seconded by Frank F. Urban, and the motion was carried.

Mr. Carroll then reported the Nominating Committee's suggestion that the following men be elected to the Board of Governors: C. M. MacGregor, Frank F. Urban, and E. E. Carroll. It was moved and seconded that the nominations be closed, and the motion was carried, and the secretary was so instructed.

The meeting was then turned over to Mr. Taylor, chairman of the By-Laws Committee. Mr. Taylor read through the suggested By-Laws for the benefit of the group, and, after much discussion regarding dues and minor changes, the By-Laws were accepted.

The new president, Mr. Kroeker, was introduced by Mr. Farnes who presented the gavel. The new vice-president, Mr. Taylor, was then introduced, as well as Secretary Heinkel, and members of the Board of Governors, Messrs. Carroll, Urban and MacGregor.

After calling for unfinished and new business, President Kroeker announced the next meeting for August 30, at which time William Goodman, Trane Co., will address the group.

The meeting was then declared adjourned.

July 13, 1939. The first meeting of Oregon Chapter, ASHVE, was called to order by Chairman B. W. Farnes with 18 members present.

It was announced that word had been received by wire that the local group of members had been granted a charter by the Council of the Society.

The question was raised as to the possibilities of allowing men applying for membership in the Society in the near future to hold charter membership in the Chapter. B. W. Moore moved that the roll of charter members be left open until August 1, 1939 and this motion was seconded by J. F. McIndoe. An oral vote was taken and it was decided unanimously to leave the rolls open until August 1, 1939.

Mr. Farnes announced that the meeting had been called for the purpose of electing a Nominating Committee so that an election of officers could be held.

A standard of procedure was adopted and the following members were elected to serve on the Nominating Committee: E. E. Carroll, chairman; J. F. McIndoe, J. P. McDermott, B. W. Moore, and B. W. Farnes. Mr. McDermott asked to be excused from the committee, because business prevented him from being available for a meeting and C. E. Heinkel was requested to serve in his stead.

The meeting was then placed in the hands of T. E. Taylor, chairman of the By-Laws Committee, who proceeded to dis-

tribute copies of the suggested By-Laws. Mr. Taylor read the By-Laws and discussion followed.

Mr. McIndoe moved that at the next meeting when the By-Laws were to be accepted that, unless discussion followed the reading of each section, that section be automatically approved. Mr. Moore seconded the motion and it was unanimously accepted by the group.

Mr. Carroll inquired regarding the advisability of incorporating and, after much discussion, the subject was postponed until a later meeting.

The business for the evening being completed, Mr. Taylor declared the meeting adjourned, according to the report of Mr. Heinkel, acting secretary.

NDHA Elects Winans

At the closing session of the 30th annual convention of the *National District Heating Association* held in New York City at the Hotel Pennsylvania, June 27-30, G. D. Winans, Detroit, was elected president for the coming year. Other officers elected were: F. L. Witsell, first vice-president; L. S. Phillips, second vice-president; and J. M. Arthur, third vice-president. Members of the Executive Committee for the coming year are J. R. McCausland, D. C. Turnbull, and R. M. Nee.

Five sessions of the convention were devoted to reports of committees and several speakers discussed each subject so that every phase was thoroughly explored.

An event of particular interest was the awarding of an honorary membership to W. J. Baldwin, Jr., past president of the Association, who has been connected with New York Steam Corp. for many years.

D. S. Boyden, Boston, past president of the Society, represented the ASHVE and gave a brief talk about the current research program. There was an interesting display of manufactured equipment which was inspected by the 200 who registered.

G. D. Winans

J. F. S. Collins, Jr.

Announcement has been made by G. D. Winans, Detroit, president of the *National District Heating Association*, that J. F. S. Collins, Jr., Pittsburgh, Pa., was appointed secretary-treasurer of the Association, with headquarters in Pittsburgh and assumed his new office August 16.

Mr. Collins is a graduate of the University of Pittsburgh and for many years has been engaged in district heating work with the Duquesne Light Co., Equitable Gas Co., and Allegheny County Steam Heating Co., affiliates of the Philadelphia Co. With this broad experience he brings to his new work a fine knowledge of the problems confronting the district heating industry. Mr. Collins has been active in the technical committee work of *NDHA* and has served as chairman of the Commercial Relations and Distribution Committees, and as a member of Operating Statistics and Membership Committees.

In the ASHVE Mr. Collins has served as president of the Pittsburgh Chapter, and is now a member of the Society's Committee on Constitution and By-Laws and the Technical Advisory Committee on Heat Requirements of Buildings.

(*First Row*)—Dr. Piettre, Paris, Mr. Raymond, London, Prof. Ernst Schmidt, Braunschweig, Prof. Keesom, Leiden, Pres. Gardner Poole, Boston, Prof. R. Plank, Karlsruhe, Arthur Woods, London, Mr. Palmieri, Rome, Dr.-Ing. Lombardi, Rome Dr.-Ing. Heiss, Karlsruhe
(*Center Row*)—Prof. Meissner, München, M. Haaser, Paris, Dr.-Ing. Nesselmann, Berlin, Miss Vobach, Miss Schaller, Dr.-Ing. Mascini, Rome, Dr. Ezer Griffiths, Teddington, Prof. Mathias, Clermont-Ferrand, and M. Chevalier, Paris
(*Back Row*)—Mr. Raymond, Jr., London, Mr. Eames, London, Prof. del Nunzio, Padua, Prof. Hausen, München, Dr.-Ing. Maiuri, London, Dr.-Ing. Altenkirch, Berlin and Dr. Saposikov, Prague

Engineers Meet at Baden-Baden and Karlsruhe

The Technical Board of the Institut International du Froid, Paris, met on July 14, 1939, at Baden-Baden, Germany, to discuss the plans for the 8th International Refrigeration Congress to be held in 1940 at Cologne and Berlin. The meeting was presided over by Gardner Poole, Boston, Mass., president of *American Society of Refrigerating Engineers* and chairman of the Technical Board of the Institute. The ASHVE was represented by Dr. Albert R. Klein, Stuttgart, a member of the Society since 1920.

On the following day the *Association of German Engineers*, who had arranged July 15 as a Refrigeration Day, met at Karlsruhe, where representatives from 10 different countries took part. During this session there were four interesting lectures given. Georges Claude, Paris, spoke on the industrial extraction of crypton and xenon from the air, and in regard to some earlier reminiscences from the air-liquefaction industry. Dr.-Ing. G. Maiuri, VDI, London, selected the new absorption machines for very low temperatures as his subject, while Dr. Ezer Griffiths, Teddington, discussed humidity measurement and control. The final lecture was on the air conditioning of mines, which was presented by Prof. Dr.-Ing. R. Plank, VDI, Karlsruhe.

Following the lectures visits of inspection were paid to the Institute of Refrigeration at the Technical University, Karlsruhe, and the Reich Institute for Food Preservation.

At the final session the preliminary program of the 8th International Congress was presented and adopted.

Australian Member on World Tour

A recent visitor to the Society's headquarters office was Jack A. Robinson, air conditioning engineer for the Australian Gas Light Co., Sydney. He has been visiting the United States and Canada for the past few weeks, and sailed for England August 12 from Quebec on the S. S. Empress of Australia.

He will visit several countries in continental Europe before returning to Australia via the Mediterranean and the Suez Canal.

His 'round the world trip commenced May 26 and he expects to return to Australia on October 2. While in the United States he visited many manufacturing plants and installations of heating and air conditioning apparatus, as well as an interesting visit to the New York World's Fair 1939.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 13 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by September 15, 1939, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
BURNS, H. J., Asst. Htg. Engr., Washington Gas Light Co., Washington, D. C.	A. E. Stack	W. H. Loving
BURTON, W. R., Engr., H. J. Sandberg, Engrg. and Sales, Portland, Ore.	P. H. Loughran, Jr.	L. B. Nye, Jr.
CLARK, R. L., Pres., Clark Asbestos Co., Cleveland, Ohio (*Reinstatement*)	J. D. Kroeker	C. E. Heinkel
DE SALES, JR., MONTEIRO, Chief Engr., Air Cond. Dept., Isnard & Co., Rio de Janeiro, Brazil	B. W. Farnes	T. E. Taylor
JONES, J. T., Power Sales Engr., Empire Dist. Elec. Co., Joplin, Mo.	L. S. Ries	J. H. Ferguson
KAHN, JR., C. R., Engr., Pabst Air Cond. Corp., New York, N. Y.	E. W. Gray	D. L. Taze
KOSTER, H. H., Asst. Prof., George Washington University, Washington, D. C.	Rudolph Kuhlmann	A. V. Hutchinson
MITCHELL, A. E., Mgr., Oil Burner and Air Cond. Dept., A. P. Woodson Co., Washington, D. C.	N. J. Botelho	R. A. Wasson
MURHARD, E. A., Pres., Muirhead and Murhard Co., Portland, Ore.	C. A. Flarsheim	J. M. Arthur, Jr.
ODUM, R. A., Supt., Htg. Dept., Grover C. Odum & Co., Daytona Beach, Fla.	A. D. Marston	L. R. Chase
RICHARDSON, L. S., Mgr., Htg. Dept., Stark-Davis Co., Portland, Ore.	H. W. Fiedler	E. E. Adams
SALE, F. B., Sales Engr., Preferred Utilities Co., Washington, D. C.	C. S. Pabst	J. G. Foster
WATSON, K. W., Mgr., Htg. Dept., Crane Co., Portland, Ore.	B. C. Cruickshanks (*ASME*)	A. F. Johnson (*ASME*)
	N. B. Ames (*AIEE*)	T. H. Urdahl
	T. H. Urdahl	A. C. Crawford
	S. L. Gregg	F. A. Leser
	J. D. Kroeker	H. B. Nielsen
	J. F. McIndoe	B. W. Farnes
	J. C. Pastor	M. C. Gillett
	H. H. Erickson	E. N. Sanbern
	A. K. Weller	T. E. Taylor
	B. W. Farnes	C. E. Heinkel
	T. H. Urdahl	I. M. Day
	W. R. Lockhart	H. M. Nelson
	J. D. Kroeker	E. C. Willey
	H. B. Nielsen	T. E. Taylor

CANDIDATES ELECTED

In the past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

ARKLEY, L. M., Head of Dept. of Mech. Engrg., Queen's University, Kingston, Ont., Can. (*Reinstatement*)

BIGOLET, LOUIS, Owner, Louis Bigolet, New York, N. Y. (*Advancement*)

DAVIES, R. H., Mgr., Airco Heating & Conditioning Pty. Ltd., Melbourne, Vic., Australia.

FUNCK, E. H., Sales Mgr., Johnson Fan & Blower Corp., Chicago, Ill. (*Advancement*)

GOODRAM, W. E., Partner, Goodram Bros., Htg. Contractors, Hamilton, Ont., Can. (*Advancement*)

HARRIGAN, E. R., Vice-Pres., Harrigan & Reid Co., Detroit, Mich. (*Reinstatement*)

KINCAIDE, M. C., Chief Engr. in charge of Air. Cond., Timken Silent Automatic Co., Detroit, Mich. (*Advancement*)

MCCLANAHAN, L. C., Dist. Mgr., Aerofin Corp., Dallas, Tex. (*Reinstatement*)

NICOLL, S. F. Air Cond. Engr., York Ice Mach'nery Corp., York, Pa.

WOODGER, H. W., Maintenance Engr., General Electric Co., Pittsfield, Mass.

ASSOCIATES

PATTERSON, J. D., House Htg. Engr., Chattanooga Gas Co., Chattanooga, Tenn.

PONDER, E. A., Mgr., Oil Burner & Warm Air Dept., A. G. Rushlight & Co., Portland, Ore.

ROSSITER, I. J., Sales Engr., American Blower Corp., Milwaukee, Wis.

SCHROEDER, W. R., Sales Engr., General Htg. Corp., Chicago, Ill.

TILTON, N. K., Vice-Pres. in charge of Sales, Mayflower-Lewis Corp., St. Paul, Minn.

JUNIORS

CAMPBELL, G. W., Engr., T. H. Urdahl, Cons. Engr., Washington, D. C.

COLBY, J. H., Sales Engr., Johnson Service Co., Boston, Mass.

COOPER, D. E., Partner, D. E. Cooper and Son, Salem, Ore.

GHOSH, B. B., Air Cond. Engr., Messrs. Refrigerators (India) Ltd., Calcutta, British India

JACKSON, W. F., Lab. Engr., American Stove Co., Lorain, Ohio

LYFORD, R. G., Sales Engr., The Powers Regulator Co., Dallas, Tex.

STUDENT

MCGOWN, JR., F. H., Student, Yale University, New Haven, Conn.